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Can He Beat This Record?

Use of an Indicator in Setting Engine Valves

By Charles Munson

SYNOPSIS—Showing the importance of proper valve setting of a compound air compressor, also the effect on the indicator diagrams with different adjustment of the valves. The saving in steam consumption with properly adjusted valves over that used with improperly set valves is also shown.

All of the accompanying indicator diagrams were taken while setting the valves on a cross-compound, two-stage air compressor. As the load was fairly constant they clearly show what effect the various adjustments have on the steam distribution in the cylinders; also, the effect proper valve adjustment has on the steam economy.

This compressor had an 18 and 30 by 32-in. steam cylinders and 26x15-in. air cylinders. The speed was about 60 r.p.m. Each cylinder had a 2⅝-in. piston rod passing through both heads. The valve-gear was a modified form of the Myers gear with riding cutoff valves. Both the main and auxiliary valves were driven by eccentrics, cast solid with each other. The cutoff could be adjusted to suit the load, while the compressor was in operation, by turning a handwheel on the cutoff valve rod. The steam pressure was 130 lb.

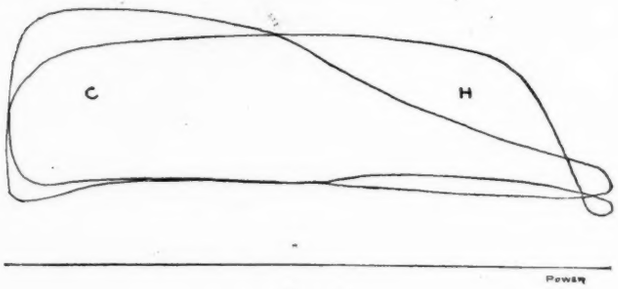


Fig. 1

In Figs. 1 and 2 are shown the condition of the valve setting in the high- and low-pressure cylinders, respectively, before any adjustments were made. Referring to Fig. 1, the head-end diagram shows very late admission and full steam pressure is not admitted to the cylinder until the piston has traveled approximately 20 per cent. of its forward stroke. This also accounts for the negative loop at the beginning of the admission line. Cutoff does not occur until nearly the end of the stroke, the auxiliary valve being so adjusted as to be useless. The crank-end diagram is better, although far from being good. The principal fault is a late admission.

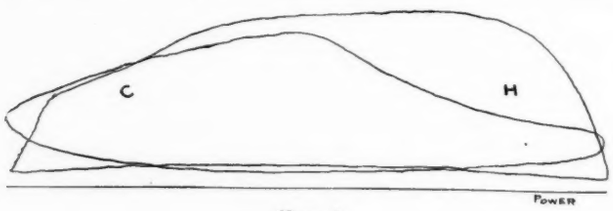


Fig. 2

Initial steam pressure on the head-end diagram is only 90 lb. per sq.in. and on the crank-end, 100 lb., although the boiler pressure was 130 lb. per sq.in. The difference in pressure in the two ends of the cylinder was due to the fact that the head-end valve did not open wide enough to permit the pressure to build up in the cylinder above 90 lb. The 30-lb. drop in pressure between the boilers and the cylinder occurred because the speed of the compressor was controlled by the throttle and not by adjusting the cutoff. Although cutoff occurs much earlier in the crank end than in the head end the indicated horsepower of both ends of the cylinder is the same. Comparing the total indicated horsepower of the high-pressure cylinder with that of the low-pressure cylinder the former is doing more work because too low a receiver pressure was being carried.

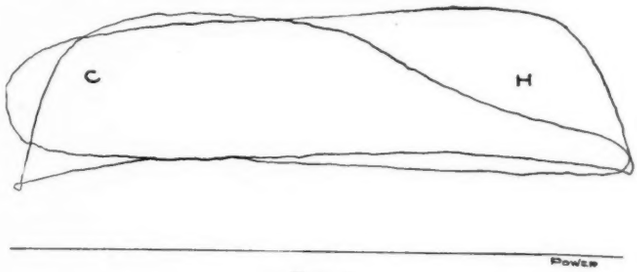


Fig. 3.

Adjustment was first made to equalize the events in the two ends of the high-pressure cylinder. The main valve rod was shortened two turns, making the head-end admission earlier and the crank-end admission later. Fig. 3 shows the result of this adjustment. The negative loop of Fig. 1 has disappeared and full admission pressure occurs at about 12 per cent. of the stroke. A small negative loop has appeared at the beginning of the admission line at the crank end and full admission pressure does not occur until approximately 15 per cent. of the stroke.

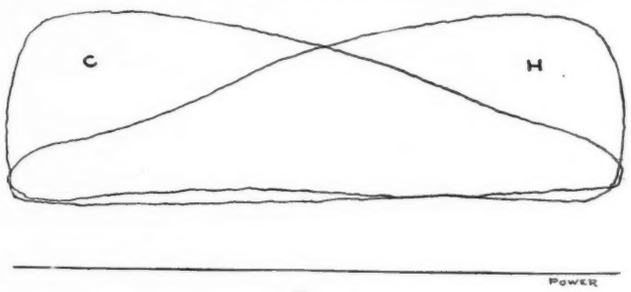


Fig. 4

Admission pressure in the head end has been raised from 90 to 95 lb. and in the crank end the initial pressure has been dropped from 100 to 90 lb. Fig. 3 also shows that the valve rod was shortened a little too much and its length was increased one-half turn before the diagrams shown in Fig. 4 were taken. To make all events in both ends of the cylinder occur earlier the eccentric was advanced about 2 in.

As both the main and cutoff eccentrics are cast in one piece, both eccentrics were advanced the same amount. The cutoff valves were then adjusted to equalize the

cutoff and the diagrams shown in Fig. 4 were taken. They show a much earlier admission and release than those in Fig. 3, and that some compression has been added. The load between the two ends has been nearly equalized and the initial pressure raised to 100 lb. on both ends. The peculiar expansion line of the crank-end diagram is due to a leaky valve, caused by changing the travel of the valve and taking the indicator diagram before it had worn down to a tight seat.

After advancing the eccentric ½ in. more the diagrams in Fig. 5 were taken. In getting these results no changes were made in the low-pressure valve setting nor was the cutoff in the high-pressure cylinder made earlier. After adjusting the low-pressure valves, the low-pressure cut-

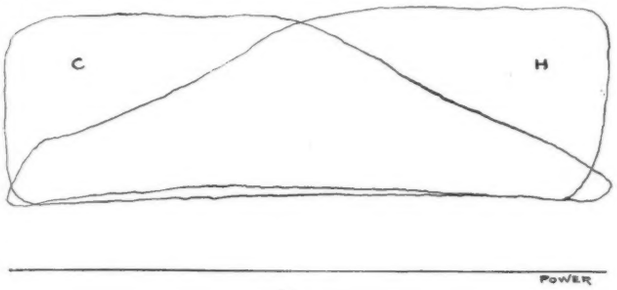


FIG. 5.

off was made earlier, and the receiver pressure was increased, the low-pressure cylinder doing a larger proportion of the work. At the same time the work done in the high-pressure cylinder was decreased. The throttle was then opened wide and the high-pressure cutoff occurred earlier, thereby keeping the speed of the machine constant.

Fig. 6 shows the results obtained. These diagrams were taken after all adjustments were made. The initial pressure has been increased to 120 lb. and the back-pressure raised to 37 lb. The indicated horsepower has been reduced from 114 to 108.91. In Fig. 2 is shown the low-pressure diagrams before adjusting the valve setting. Admission and release are both late and the crank-end cutoff is much earlier than in the head end. The low-pressure valve rod was shortened to equalize events, and the eccentrics advanced about 1 in. to make all events occur earlier. After adjusting the cutoff valves, diagrams were taken, as shown in Fig. 7.

Admission and release are still late; more so on the head end than on the crank end. Cutoff is also later

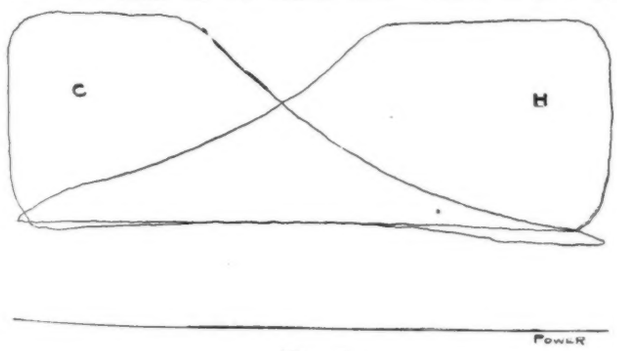


FIG. 6

on the head end than on the crank end. The valve rod was shortened about three-quarters of a turn and the eccentric advanced ½ in. more. The diagrams shown in Fig. 8 were then taken.

These adjustments have raised the initial pressure about 2 lb., but more work is being done in the high-pressure cylinder than in the low. The cutoff valves were further adjusted and the cutoff was made to occur earlier until the receiver pressure was raised to 37 lb. per sq.in. Diagrams shown in Fig. 9 were then obtained.

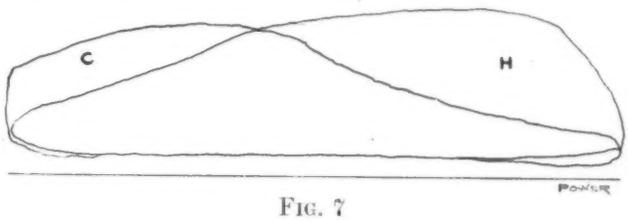


FIG. 7

The speed of the compressor has been reduced from 60 to 55 r.p.m., although the total indicated horsepower is practically the same as at first. Before the diagrams shown in Figs. 6 and 9 were taken, the air valves were cleaned and new piston rings put in both air pistons. This so reduced the leakage that the speed could be dropped 5 r.p.m. and still carry the load.

In order to show more clearly the effect of these various changes, Figs. 1, 2, 6 and 9 have been combined in Fig. 10. The clearance was found to be approximately 7 per cent. for each cylinder. The cylinder ratio is 30^2 to 18^2 or 2.77. The total volume of the high-pressure cylinder referred to the low-pressure cylinder is

$$(1 + 0.07) \div 2.77 = 0.386$$

and

$$0.386 - 0.07 = 0.316$$

In other words, the end of the high-pressure diagrams should be plotted at 31.6 per cent. of the low-pressure diagrams. Dividing the high-pressure volume by the cylinder ratio 2.77, and drawing all four diagrams to the

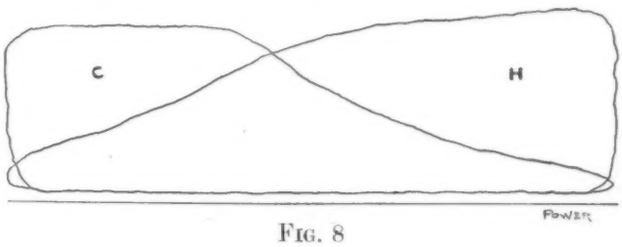


FIG. 8

same pressure scale, Fig. 10 is obtained. Dotted lines show the diagrams taken before setting the valves and the full lines indicate those taken after all adjustments had been made.

Particular attention is called to the increase in initial pressure in the high-pressure diagrams, decrease in the drop in pressure through the receiver, higher receiver pressure and lower release pressure in the low-pressure cylinder. This last point shows that the expanded steam has been carried further and consequently better economy has been obtained.

The following figures will give an approximation of the saving which has resulted by having the valves properly set. At point *A*, Fig. 10, which is 90 per cent. the head-end stroke, and 90 per cent. of the volume of the low-pressure cylinder plus the clearance, 7 per cent., makes a total of 97 per cent. if the low-pressure cylinder is filled with steam at 13.3 lb. gage or 28 lb. absolute. This steam contains more or less moisture, due to cylinder condensation. However, it is impossible to determine what

the percentage of moisture is from the ordinary indicator diagram, so it is assumed to be dry at this point. This assumption will not affect the actual saving as the same assumption is made in calculating the steam consumption after setting the valves. It will make the steam consumption per indicated horsepower lower than actual, however.

From the steam tables it is found that the weight of a cubic foot of dry steam at 28 lb. per sq.in. absolute is 0.068 lb. The volume of the low-pressure cylinder is

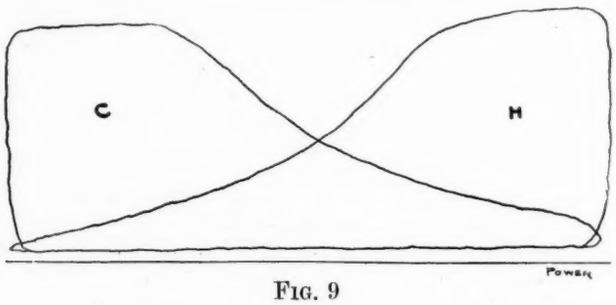


FIG. 9

12.99 cu.ft. after deducting the volume of the piston rod, which is 2⅝ in. in diameter there remains,

$$0.97 \times 12.99 = 12.6 \; cu.ft.$$

The total weight of steam in the head end of the cylinder up to point *A* is,

$$12.6 \times 0.068 = 0.8568 \; lb.$$

at the beginning of the stroke *B*, the clearance space is filled with steam at 1.5 lb. gage or 16.2 lb. absolute. The weight per cubic foot of dry steam at this pressure is 0.041 lb. and the volume of the clearance is 7 per cent. of 12.99 or 0.91 cu.ft. The total weight of steam in the clearance then is

$$0.041 \times 0.91 = 0.0373 \; lb.$$

This steam is already in the cylinder when the admission valve opens so that the only steam admitted at the head-end stroke is the difference between that shown at *A* and *B* or

$$0.8568 - 0.0373 = 0.8195 \; lb.$$

In a like manner the pressure at *C* is 23.7 lb. absolute,

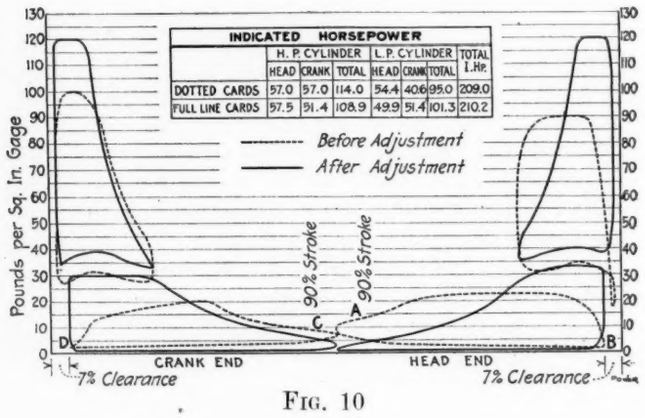


FIG. 10

and the weight per cu.ft. is 0.058 lb. At *D* the pressure is 17 lb. absolute and the weight 0.043 lb. per cu.ft. Therefore, the steam used at the crank-end stroke is:

$$(12.6 \times 0.058) - (0.91 \times 0.043) = 0.6917 \; lb.$$

The total steam per revolution

$$0.8195 + 0.6917 = 1.5112 \; lb.$$

Dry steam per hour equals

$$1.5112 \; lb. \times 60 \; r.p.m. \times 60 \; min. = 5440 \; lb.$$

The total indicated horsepower before setting the valves (diagrams Figs. 1 and 2) were 209. The dry steam per indicated horsepower per hour equals

$$5440 \div 209 = 26.03 \; lb.$$

In the same way the dry-steam consumption after setting the valves was calculated to be 3688 lb. per hr., the indicated horsepower 210, and the dry steam per indicated horsepower 1755 lb. At the plant in question 8.5 lb. of water were evaporated per pound of coal. The compressor runs for about 10 hr. per day for 300 days per year. The steam saved per hour is

$$5440 - 3688 = 1752 \; lb.$$

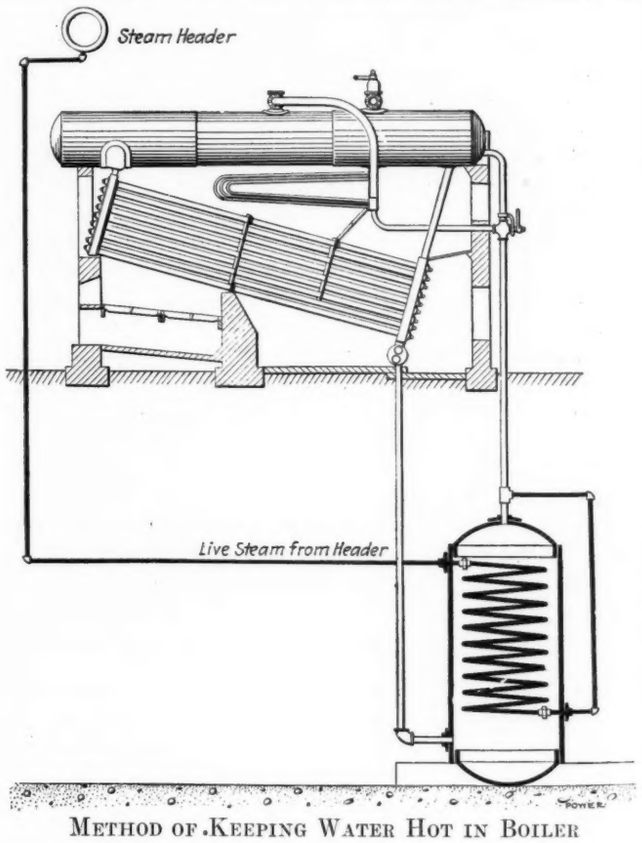
and

$$1752 \div 8.5 = 206 \; lb. \; of \; coal \; per \; hr.$$

In other words, during a day of ten hours about a ton of coal was saved as a result of properly setting the valves on this compressor.

Quick Firing of Standby Boilers

A novel method of quick firing for boilers is described in the *Electrical World* for Dec. 6, with reference to the Ashley Street steam plant of the Union Electric Light & Power Co., of St. Louis. This company purchases electricity from the Mississippi River Power Co., at Keokuk, Iowa, to supply part of its system, but to guard against interruptions, should the hydro-electric power fail, the



METHOD OF KEEPING WATER HOT IN BOILER

steam station is operated in such a manner as to be able to take over the entire load in a few minutes in an emergency.

This state of readiness is accomplished by keeping the 20 extra boilers full of water at 212 deg. and with fuel ready to ignite on the grates. The grate is first covered with a high-grade nut coal to a depth of about 7 in. and three rows of oily rags are then placed in cross trenches. Another row of rags extends down the middle to carry

along the flame to all parts of the grate. On top of these rows of rags is laid kindling wood soaked in kerosene, while a few gallons of fuel oil are sprinkled over the entire bed of coal. This produces a hot fire very quickly and the best time made thus far in getting a boiler on the line, starting from zero-gage pressure, has been 16 min. This is a much shorter time than is required to get a boiler up to pressure from a banked fire.

The method of keeping water hot in the emergency boiler is shown in the illustration. The common blowoff header connecting the boilers in service was connected to the lower part of one of the two blowoff tanks in the basement. This header serves to bring the cool water from the lowest part of each boiler. In the tank is installed a coil through which the waste steam from one of the large steam headers is caused to flow, thus serving also to drain the header. The steam heats the water in the tank, sets up a circulation and delivers hot water to the steam drum of each boiler. The water is thus kept at practically the boiling point at a coal expenditure of only 12.5 lb. of coal per boiler per day as compared with 3100 lb. with banked fires.

Air-Compressor Testing

By W. C. Lancaster

SYNOPSIS—Compressor efficiencies, data from test of a motor-driven two-stage compressor and the computations necessary to obtain the results.

Competition among air-compressor builders has been so keen in recent years that the would-be purchaser is often perplexed by the mass of data and technical figures presented by the manufacturer. Confusion lies chiefly in the question of efficiency. The mere statement that an air compressor has an efficiency of so many per cent. is not sufficiently definite. To be clear, the kind of efficiency that is meant must be stated since there are several different kinds all applicable to air compressors.

"Volumetric efficiency," or "delivery efficiency" as it is sometimes called, is the ratio of the volume of free air at intake temperature, actually delivered by the compressor per unit of time, to the displacement volume of the low-pressure cylinder in that time.

"Mechanical efficiency" represents only the loss due to the friction of the machine and heating of the air and jacket water. It may or may not include the belt or gear loss, if either of these drives is used, or it may include the friction loss of the steam engine if the compressor is directly connected to one.

"Compression efficiency" is a term sometimes used to express how close the actual compression line of a compressor comes to the isothermal line. If the isothermal line is drawn on the indicator card from the beginning of the stroke to the theoretical delivery line, the compression efficiency is the area under the isothermal and theoretical delivery lines divided by the actual area of the card.

"Electrical efficiency," of course, has nothing to do with the air compressor itself, but applies to the losses in the electric motor if used to drive the compressor.

All these various efficiencies are of much interest to the mechanical engineers who design air compressors and to the manufacturers who build them, but to the man who pays the bills for coal or electricity to run his plant they are usually only confusing. What he is really interested in is the number of cubic feet of air per minute actually delivered by his compressor per horsepower input. This represents the true economy of the compressor. It can be stated somewhat differently, as, for instance, "the kilowatt-hours input per 1000 cu.ft. of free air delivered" or "the cubic feet of air per pound of steam," or "the kilowatt input per 100 cu.ft. of free air per min. delivered." But all these are merely different ways of stating the relation between the amount of air actually compressed and pumped out by the compressor and the energy required to accomplish it. This is what the air-compressor operator really wants to know about a compressor before he makes the purchase, as upon this relation depends the amount of money he will have to pay out year after year for power to operate it.

After a compressor has been installed it is therefore desirable to run a test to determine exactly what the economy of the compressor is, both as a check on the manufacturers' guarantee and for the information of the owner.

A method of making such a test that has been found satisfactory by the writer will first be described, and then as an illustration an actual test will be worked out in detail. So much has already appeared in the technical press on the subject of steam-driven compressors that this article will be confined to those driven by electricity.

The two most important measurements to be made are:

a. The number of cubic feet of free air actually delivered by the air compressor.

b. The power input into the motor that drives the compressor.

Other measurements that will serve as a check on the above and give additional information about the compressor are:

c. (1) Indicator cards, (2) intercooler air pressure, (3) intercooler air temperature, entering and leaving, (4) intercooler water temperature, entering and leaving, (5) high- and low-pressure jacket-water temperature, entering and leaving, (6) amount of cooling water used in each cylinder jacket and intercooler.

Taking up these measurements in the order named, it has been found that the quantity of air flowing through an orifice varies directly as the area of the orifice and the absolute pressure and inversely as the square root of the absolute temperature. For pressures in the reservoir greater than twice the atmospheric pressure this relation is expressed mathematically in Fliegner's formula as follows:

$$G = 0.53\, F \frac{P}{\sqrt{T}} \qquad (1)$$

where

G = Flow of air in pounds per second;
F = Total area of all the orifices in square inches;
P = Absolute pressure in pounds per square inch in the orifice chamber;

T = Absolute temperature Fahrenheit of the compressed air in the orifice chamber.

For large air compressors a number of orifices are required. The orifices should be near the compressor and a pipe of large diameter and as straight as possible used to connect the discharge opening to the orifice manifold. The orifices are accurately ground in hardened-steel plates of from $\frac{3}{8}$ to $\frac{1}{2}$ in. thickness with the inside edges of the orifices rounded to a radius of $\frac{1}{16}$ in. less than the thickness of the plate, leaving $\frac{1}{16}$ in. of the hole straight. The shape of these orifices is illustrated in Fig. 1 and the method of assembling them with the valves and manifold is shown in Fig. 2. These plates with different sizes of orifices are placed in the manifold and the valves opened and closed until just the right orifices are found which will hold the pressure at whatever value it is desired to make the test.

A thermometer is inserted in the thermometer well provided in the orifice manifold and a pressure gage connected. The compressor is run until all parts are heated

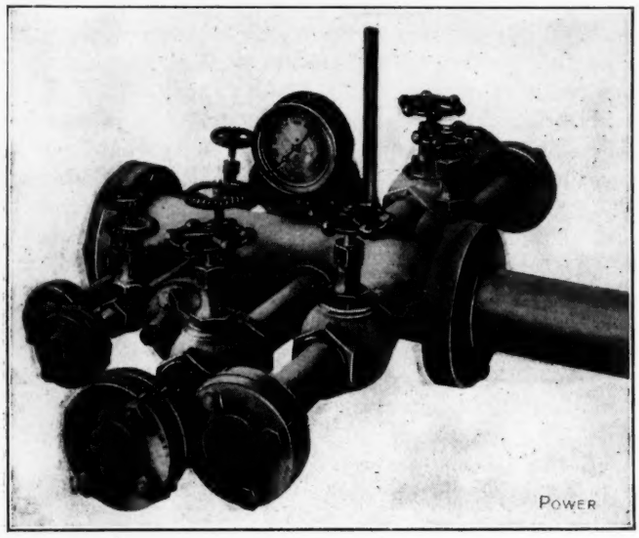


FIG. 2. ORIFICE MANIFOLD

up and then readings are taken at short and regular intervals until the pressures and temperatures become constant. The amount of air discharged can then be figured by the use of equation (1). The writer believes that this method can be relied upon to give results accurate within about 2 per cent.

The power input to the motor is obtained by inserting suitable electrical-measuring instruments into the motor circuit. If the motor is a direct-current machine, an indicating voltmeter and ammeter will be required. It will usually be found that two ammeters will be needed, one to read the current at full load and the other for the current when the compressor is "on a vacuum, as in this case the current is so small that the ammeter large enough to show the current at full load would read too low on the scale for accuracy. The power input is the product of the current and voltage. Several successive readings should be taken at the same time that the readings of the pressure gage and thermometer on the orifice manifold are read, and these readings averaged.

For a three-phase motor, either a three-phase wattmeter or two single-phase wattmeters can be used. If the latter, the power is the algebraic sum of the wattmeter readings. In either case, it may be necessary on account of large currents or high voltage to insert either current or voltage transformers, or both, between the motor circuit and the instrument. If this is done the meter readings must be multiplied by the ratio of the transformer, or by the product of the two ratios if both current and potential transformers are used.

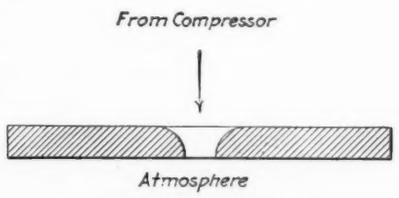


FIG. 1- ORIFICE

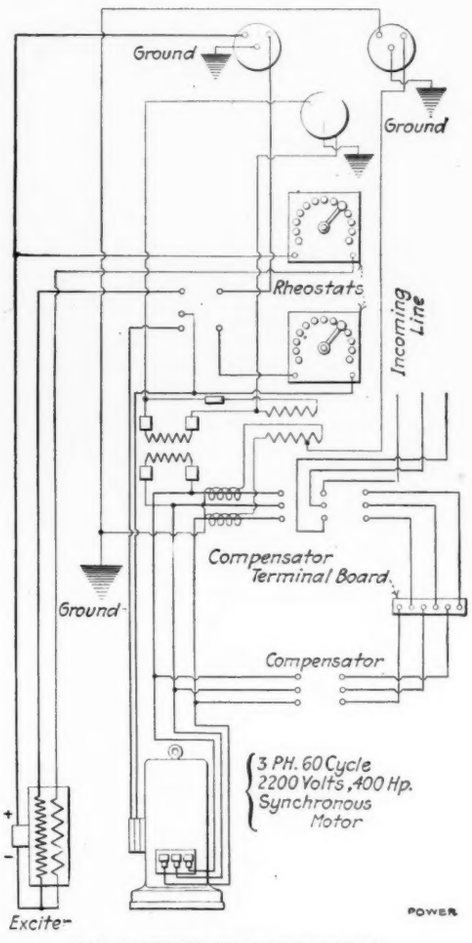


FIG. 3-MOTOR WIRING DIAGRAM

For two-phase circuits, two wattmeters are used, the power being the arithmetical sum of the wattmeter readings multiplied by the transformer ratios.

Indicator diagrams should be taken on every test and if possible four indicators should be used so as to obtain simultaneous cards from each end of each cylinder. They are useful to show whether the valves were properly set during the test, and also to give the compression and mechanical efficiencies.

The method of figuring the compression efficiency has already been described. The overall mechanical efficiency is the ratio of the indicated horsepower, obtained in the usual way from the indicator diagrams, to the horsepower input into the motor. If it is desired to obtain the mechanical efficiency of the compressor only, it can be done quite accurately in the case of direct-connected com-

pressors by making due allowance for losses in the motor from the motor builder's efficiency curve, but for belt-driven compressors it is difficult to separate the belt loss from the friction losses of the machine.

Readings *c*, from 2 to 6 inclusive, should all be taken, as they show the performance of the intercooler and cylinder jackets. The amount of cooling water required is an important item in the operating expense of an air-compressor plant. While usually small compared with the cost of power to drive the compressors, it is, nevertheless, quite appreciable, whether the water be purchased from city mains or pumped from a near-by source of supply, and should always be carefully reckoned in considering the economy of an air compressor.

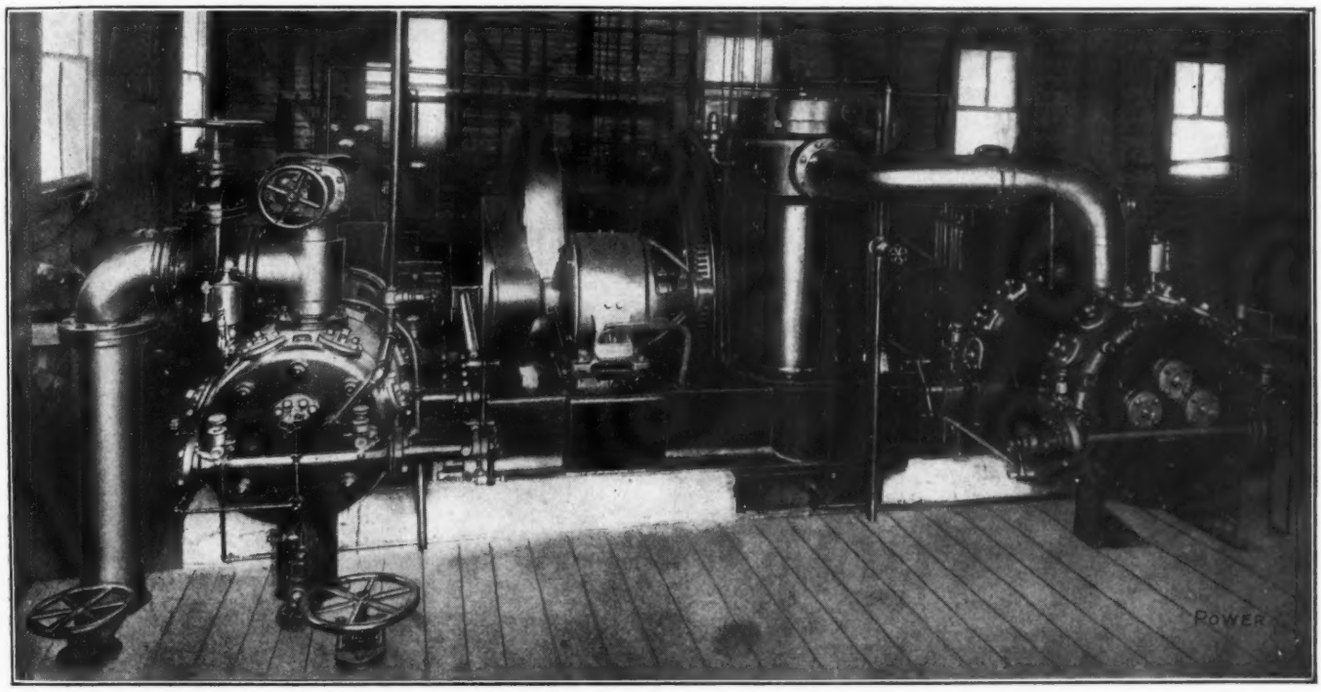


FIG. 4. MOTOR-DRIVEN TWO-STAGE AIR COMPRESSOR, 2100 CU.FT.

TEST OF MOTOR-DRIVEN AIR COMPRESSOR

The test run that will be recorded and then worked out as an example of the method described was made on a 26x15½x18-in. Sullivan two-stage compressor direct connected to a 400-hp., three-phase, 60-cycle, synchronous motor mounted on the crankshaft. The voltage is 2200 in the stationary armature and the revolving field is excited by current from a small generator belted to the flywheel. A wiring diagram of the motor is given in Fig. 3. There are two Corliss inlet valves on each cylinder, moved by eccentrics on the crankshaft. The air is discharged through poppet valves. There is an unloading valve on the inlet of the low-pressure cylinder, operated by air piped from the receiver and so adjusted that it will operate within a range of pressure of 5 lb. This valve is either fully open or tightly closed, so that the compressor is operating either at full load or at no load.

This air compressor is one of a number installed in the compressor plants for the construction of the tunnel under Mount Royal, Montreal, now being built for the Canadian Northern Ry. A view of the machine as it is installed is given in Fig. 4.

CALCULATIONS

With a low-pressure cylinder of 26x18 in., the displacement per revolution is 19,021.14 cu.in. The speed in revolutions per minute is

$$(157{,}665 - 153{,}032) \div 25 = 185.3$$

The displacement equals

$$\frac{19{,}021.14 \times 185.3}{1728} = 2039.7 \textit{ cu.ft. per min.}$$

The area of the orifices is as follows:

One ¾-in. orifice, sq.in.	0.3068
Three ½-in. orifices, sq.in.	0.58905
One ¼-in. orifice, sq.in.	0.04908
Total, sq.in.	0.94493

The absolute pressure of the atmosphere, with a barometer of 29.9395 in. equals

$$29.9395 \times 0.4908 = 14.694 \textit{ lb. per sq.in.}$$

Adding to this 103 lb., which from the log is the pressure at the orifice, gives 117.694 lb. per sq.in. abs. as the pressure in the orifice chamber.

The temperature in the orifice chamber from the log is 227 deg. F. Adding 460 to find the absolute temperature gives 687, and the square root of 687 equals 26.216.

At the intake temperature of 59 deg. F. the weight of 1 cu.ft. of air is

$$W = \frac{1.325 \times B}{460 + t} = \frac{1.325 \times 29.9395}{460 + 59} = 0.07643 \textit{ lb.}$$

From Fliegner's formula

$$G = 0.53 F \frac{P}{\sqrt{T}}$$

$$G = \frac{0.53 \times 0.94493 \times 117.694}{26.216} = 2.248 \textit{ lb. per sec.}$$

$$\frac{2.248 \times 60}{0.07643} = 1764.75 \textit{ cu.ft. per min. of free air at 59 deg. F.}$$

The volumetric or delivery efficiency equals

$$1764.75 \div 2039.7 = 86.5 \textit{ per cent.}$$

A method often used to figure the volumetric efficiency is to divide the length of the line *ab*, Fig. 5, by the length of the atmospheric line *ac*. This method, however, gives too high a value as it does not allow for slippage at the discharge valves and leakage past the piston. It might more properly be called "intake efficiency" when figured this way.

For the test considered, the volumetric efficiency figured in this manner works out as follows: For the crank end $ab \div ac = 96.9$ per cent., and for the head end, 98.8 per cent. The average of these is 97.85 per cent., which, when compared with the value 86.5 per cent. found above, illustrates clearly what a widely different result may be obtained by the indicator-diagram method.

At 111 lb. per sq.in. terminal pressure, the average of the last three readings, the power input was 316 kw. Reducing this to 100 lb. per sq.in. terminal pressure, on the assumption that the power is proportional to that required for adiabatic compression (which is a sufficiently close approximation),

$$316 \times \frac{15.3}{16.15} = 299.4 \; kw. = 401.8 \; hp.$$

Kw. input per 100 cu.ft. per min., free air at 100-lb. gage output =16.96
Hp. input per 100 cu.ft. per min., free air at 100-lb. gage output =22.75
Cu.ft. per min., free air output at 100 lb. gage, per kw.-hr.. input = 4.395
Kw.-hr. per 1000 cu.ft. free air..... 2.83
Cu.ft. per min., free air output at 100 lb. gage, per kw.-hr. input. =353.5
Power input at no load with the compressor operating at full speed, but with the intake cut off and, therefore, pumping no air, kw.....38

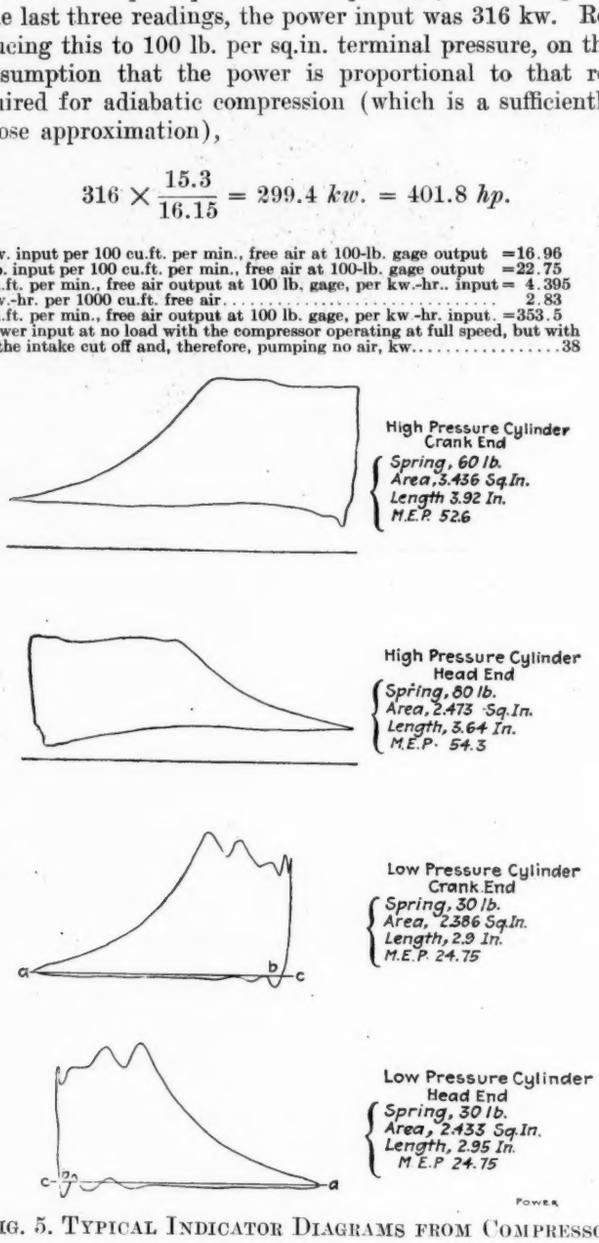


FIG. 5. TYPICAL INDICATOR DIAGRAMS FROM COMPRESSOR

Typical indicator diagrams, taken during the test, are shown in Fig. 5. The indicated horsepower of the low-pressure cylinder equals

$$\frac{24.75 \times 530.93 \times 1.5 \times 371}{33{,}000} = 221.55$$

The indicated horsepower of the high-pressure cylinder equals

$$\frac{53.45 \times 188.69 \times 1.5 \times 371}{33{,}000} = 170.08$$

Adding gives 391.6 i.hp. The total input = 316 kw. = 423.6 hp. Then the mechanical efficiency equals

$$\frac{391.6}{423.6} = 92.5 \; per \; cent.$$

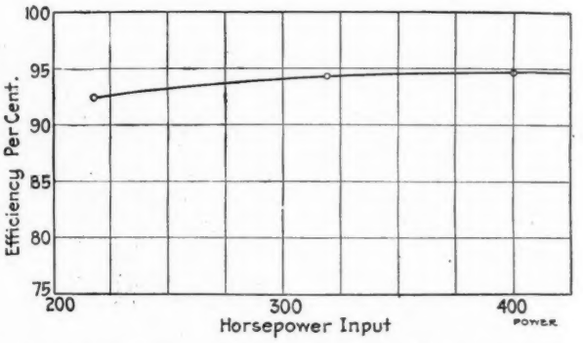


FIG. 6. EFFICIENCY CURVE OF 400-HP. SYNCHRONOUS MOTOR

From the efficiency curve of the motor, furnished by the motor manufacturer, Fig. 6, it will be noticed that the electrical or motor efficiency at 423.6 hp. input is 94.6 per cent.

Since the overall efficiency has been found to be 92.5 per cent., the mechanical efficiency of the compressor exclusive of the motor is

$$0.925 \div 0.946 = 0.977 = 97.7 \; per \; cent.$$

SUMMARY OF TEST RESULTS

Hp. input per 100 cu.ft. per min. of free air at 100-lb. gage output	22.75
Cu.ft. per min., free air output at 100-lb. gage per hp. input	4.395
Kw.-hr. per 1000 cu.ft. free air	2.83
Power input at no load, kw	38.00
Overall mechanical efficiency, including motor, per cent	92.5
Mechanical efficiency exclusive of motor, per cent	97.7
Volumetric or delivery efficiency, per cent	86.5
Cooling water during test, cu.ft. water per 1000 cu.ft. air	1.00

Interviewing a Grate Salesman

BY R. O. RICHARDS

"Why do my grates deteriorate faster at the front than at the back end of a furnace?" the chief inquired in the course of conversation with the visiting grate salesman.

"Because," the salesman replied, tipping his hat to the back of his head, "in ordinary firing more pounds of coal per square foot of grate surface are consumed there, while, relative to the rate of combustion, less air—the grate-cooling factor—passes through the air spaces. The volume of air which passes through the body of the fire, immediately after a fresh charge of fuel is fed to the furnace, is insufficient for complete combustion, and the auxiliary supply admitted through the grating of the furnace doors, is so limited in volume that only the fuel immediately back of the doors is benefited. Consequently, the front part of the fire reaches incandescence sooner than any other part of the furnace, with the possible exception of narrow strips placed next to the heated side walls.

"Air admitted above the fuel is not a grate-cooling factor, and the air supplied through the ashpit doors is retarded somewhat from reaching the front quarter of

the grates because of the sharp deflection in the direction of its flow.

It is not difficult to conceive that a strip of grate in the front of the furnace, varying in width from 1 to 2 ft. is subject to comparatively high temperatures, and therefore deteriorates more rapidly than the rest of the grate. For this reason I recommend either slightly closer air spaces in the rear grates, or more cooling surface, greater depth of bar, in the front grates.

"Although holding that the lack of sufficient cooling surface is the chief cause, I am of the opinion that an unequal and slovenly manner of firing is also a fruitful

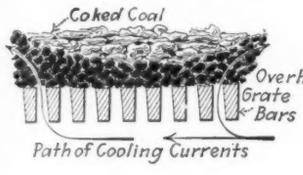


FIG. 1. AIR PATH THROUGH FUEL

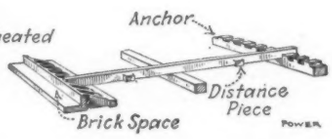


FIG. 2. CROSS-BAR GRATE SUPPORTS

source of grate deterioration. When a hole is burnt in the body of the fuel most of the air will pass through this point of least resistance, and consequently the rest of the grate will suffer from the heat. Especially is this true with heavy fires, light steam demand and partly open damper.

"If the fireman, instead of leveling the fire, fills the hole with green coal, the draft will be almost checked there, and this green spot will soon suffer from excessive heat by the coking of the coal on top, and its ignition by convection near the grate. Fig. 1 makes this point clear."

"Now, tell me," said the chief, "why, when a grate-bar warps, it burns and breaks sooner than a bar that has not warped, but is otherwise not in as good condition. What is the reason?"

"That," replied the salesman, "occurs most frequently to a bar that has arched upward than to one that has sagged. One of the principal factors of grate economy is the maintenance of an even surface—one without irregularities in either the bars or the finger-pieces. In general, the bars wear evenly relative to one another, and, if they be so maintained, good service may be expected, though they may have sagged considerably at their mid length. But when they are disturbed and replaced without regard to their former evenness of surface, you may be sure that they will soon be destroyed.

"It is as important to number the bars of a grate when it is being removed as to mark the parts when disjoining any apparatus in the engine room. If this is not done the grate surface will be uneven (Fig. 3) when replaced and from the moment the furnace is lighted, the fire on the one hand, and the fireman's slice-bar on the other, will contend which shall level the most projections in the shortest time."

"Which is the better, a thick bar or a thin one?" asked the chief, as he stabbed a blotter with his new fountain pen. "Many engineers infer that since less grate surface is in contact with the fire, a thin bar is preferable, but reliable tests have proved that the grate receives heat as much from radiation through the interstices, as from direct contact with the burning fuel. Consequently, there being less material and only the same amounts of cooling surface, a thin bar must become many degrees hotter than a thick one, and, not having the rigidity of the latter, will soon bend sidewise, so that two or more come in contact, causing wasteful enlargement of the air space on one side and serious diminution of the cooling surface on the other.

"Other considerations favorable to the use of thick grate bars are that they better sustain a layer of protective ash, and maintain the line of fire farther removed from their own surface.

"What factors govern the depth of a bar," was the chief's next inquiry, to which the salesman replied without hesitation:

"The length of the bar, the amount of fuel burned per square foot per hour, and the width of the air spaces. The latter consideration is rarely taken into account in grate design, but is a factor that materially influences the temperature of the grate. Grates between which are wide air spaces are subject to more radiation of heat than those having narrow air spaces between them and they should show more cooling surface or a greater depth of metal in the bar.

"In this country cast-iron grates have little or no competition; but in Europe steel and wrought-iron grates are used in plants that consider length of service and not initial cost, as the true measure of grate economy. The most common form is a series of straight bars extending unbroken from the front arches to the bridge-wall, and supported at the center by a girder. The bars are generally forged 4 in. deep, and about three-quarters of an inch thick. They are tightly packed latitudinally by distance pieces, but are free to expand lengthwise, being anchored near the bridge-wall. (See Fig. 2.) Their chief advantage is, that warped bars may be straightened and impaired bars welded and reforged. They have a smooth

FIG. 3. UNEVEN GRATE

FIG. 4. SLICE BAR

surface and the fires are more easily barred. The fire end of the slice bar is forced so that the lower lip slides between two bars and cuts all clinkers that may adhere to the sides." (See Fig. 4.)

Just then the whistle blew for dinner and the salesman, losing all interest in talking grates, caught a trolley headed for the hotel dining room.

Pumping against 2250-Lb. Pressure*

Water used by the rack-and-cog-type locomotives that haul cars to the top of Mount Washington, N. H., take water from a pipe along the track instead of carrying it in a tender.

The water is pumped to the top of the mountain by duplex pumps near the base. The pressure pumped against is 2250 lb. The steam cylinders of the pump are 16 in. in diameter and the water cylinders are 3 in. in diameter. The stroke is 18 in. The discharge pipe is 2 in. in diameter and ½ in. thick for half its length and ⅜ in. for the remainder. The intake pressure is 80 lb., furnished by a dam in the Ammonoosuc River. The pumps are located 2540 ft. above sea level and pump water to the summit at an elevation of 6280 ft.

*Excerpts from "Engineering Record."

Electrical Department

Centralized Control System for Panama Locks

The operation of the Panama Canal locks, terminals and auxiliary equipment utilizes electrical energy throughout, with the present exception of the Panama Railroad, the electrification of which is under contemplation.

Generation and Distribution

Power for the operation of the locks, towing locomotives, lights, etc., is to be furnished by a 7500-kv.-a., 2200-volt hydro-electric plant at the Gatun Dam; the steam plant at Miraflores, which supplied power for construction work, will serve as an emergency plant. These will supply, through 2200/44,000-volt transformers, two parallel, three-phase, 44,000-volt transmission lines extending across the Isthmus from Balboa to Cristobal, where substations step down to 2200 volts. The electricity at these points is used for operating the coal-handling plants, the machine shops, dry docks and for general lighting.

Thirty-six 2200/240-volt substations supply the three sets of locks, Gatun, Pedro Miguel and Miraflores, with power and light. In addition there are three 2200/220/110-volt transformer stations for the control boards at the locks.

The following list gives an idea of the number and sizes of motors to be controlled in operating the lock machinery:

Machines and Operation	Motors, Each Machine and Hp.	Number of Motors: Gatun	Pedro Miguel	Miraflores	Total Hp.
Miter gate, moving, each leaf	1-25	40	24	28	2,300
Miter gate, miter forcing	1-7	20	12	14	322
Fender chain, main pump	1-70	16	16	16	3,360
Fender chain, operating valve	1-½	16	16	16	24
Rising stem gate valve	1-40	56	24	36	4,640
Cylindrical valve	1-7	60	20	40	840
Guard valve	1-25	6	6	6	450
Auxiliary culvert valve	1-7	4	4	4	84
Total		218	122	160	12,020

This does not include a number of smaller motors, such as those for the spillway gates, the hand rails on the miter gates and the sump pumps. The motors are started through contactor panels located near them, but controlled from the central control house. The smaller motors, including those for cylindrical valves, auxiliary culvert valves and miter forcing, are started by being thrown directly on the line. Two double-pole contactors are used, one for forward and one for reverse. In the case of larger motors for miter-gate moving, rising-stem valves and guard valves, a starting point with resistance in two legs of the three-phase circuit is provided.

In all cases the contactors are operated from the control boards, to be described later, by three wires, one for forward, one for reverse and a common return. In the case of panels having a starting point, the period during which the motor remains on the resistance is automatically controlled by a dashpot, so that the starting operation at the control house is the same, simply energizing a forward or reverse wire as the case may be.

Location and Operation of Lock Machinery

From an operating standpoint the machinery was placed below the coping of the lock walls, thus affording a clear space for maneuvering ships and protecting the apparatus from the weather without erecting numerous houses.

The miter gates consist of two massive leaves pivoted on the lock walls and operating independently of each other. A pair of gates is located where each change of level occurs and divides the locks into 1000-ft. chambers. In addition at the lake and ocean ends are duplicate pairs serving as guard gates. To handle the vessels of various sizes with the minimum use of water, similar miter gates are placed in intermediate positions dividing the 1000-ft. locks into two compartments.

On the top of all miter gates a foot walk with hand rails is provided. When the gates are open and in the recesses in the lock walls, these hand rails would interfere with the passing of the towing locomotives; therefore, they are raised and lowered by a motor under the foot walk, controlled from the lock wall by a foot push controller. If the hand rails happen to be left up and the gates are opened by the operator in the control house, they will be lowered automatically.

Chain fenders are stretched across the canal in front of all miter gates, which can be exposed to the upper-lock level and also in front of the guard gates at the lower end. These are maintained taut when the gates behind are closed, and are lowered when the gates are opened for the passage of a ship. The chains are raised and lowered by a method similar to that followed in hydraulic elevators, with the additional feature that if a ship approaches the gates at a dangerous speed and rams into the chain, the latter is paid out in such a way as to gradually stop the ship before it reaches the gates.

The filling and emptying of the locks is accomplished by three culverts, one in the middle wall and one in each side wall, the flow of water being controlled by rising-stem valves. These are located in the culverts at points opposite each end of each lock so that the culvert can be shut off at any desired point for filling a lock with water from above, or upstream, or for emptying it by allowing it to flow out and down to the next lock. Lateral culverts conduct the water from the main culverts, under the lock chambers, and up through openings in the lock floors.

The rising-stem valves are installed in pairs, and each pair is in duplicate; also each culvert is divided into two parallel halves by a vertical wall, which reduces the size of each valve and makes it more easily operated. One pair of duplicates is left open as an emergency or reverse pair. The culvert in the middle wall must serve the locks on both sides and to control this feature cylindrical valves are placed in the lateral culverts that branch out on each side.

Centralized Control and Indicating System

As the flight of locks at Gatun, for instance, extends over approximately 6200 ft., and, the principal operating

machines are distributed over a distance of about 4000 ft., it will be seen that a centralized electrical control is essential.

The control boards are installed in control houses (see Fig. 3), located on the middle walls at points which afford the best view of the locks, although this view is not depended on to know the position of the gates or other apparatus, as there are indicators on the control boards. These boards are operating miniatures of the locks themselves, and are arranged with indicating devices which show the position of valves, lock gates, chains and water levels, and with the exception of such machinery as needs only an open or closed indication, the indications will be synchronous with the movement of the lock machinery.

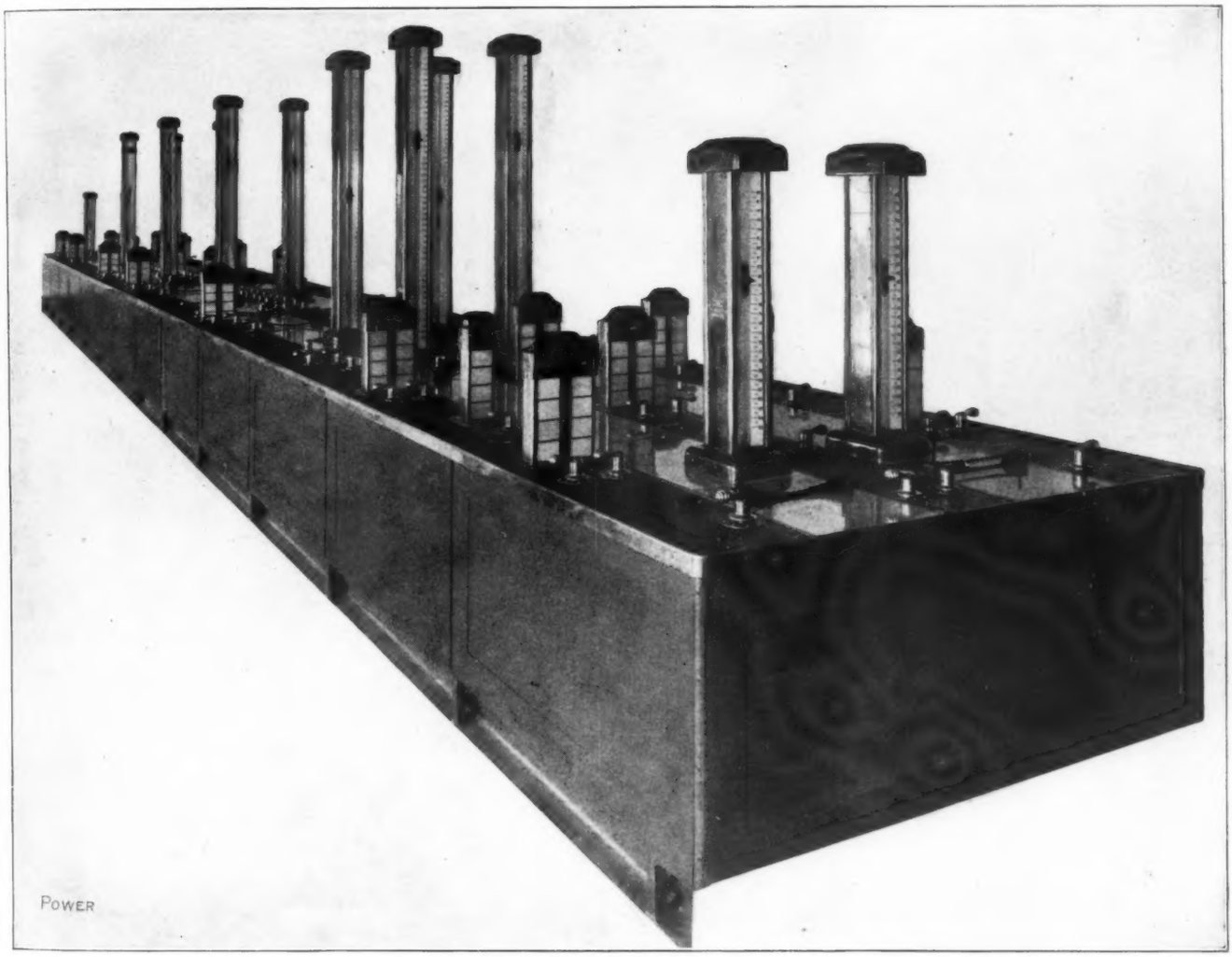


FIG. 1. CONTROL BOARD FOR MIRAFLORES LOCKS

The control boards (Fig. 1), are of flat-top benchboard type, 32 in. high by 54 in. wide, built in sections, with total lengths as follows:

Gatun	64 ft.
Pedro Miguel	36 ft.
Miraflores	52 ft.

The side and center walls of the locks are represented by cast-iron plates and the water in the locks by blue Vermont marble slabs.

The control-switch handles are mounted above the surface of the board and operate through an angle of 90 deg. They are provided with nameplates for the "open" and "closed" positions. The space immediately below the flat top of the control board is occupied by the contact fingers of the control switches, mounted on the operating shaft, synchronous receivers and their cable connections. The receivers, transmitters and lamps are operated at 110 volts, while the control circuits are at 220 volts, both using 25-cycle alternating current.

The synchronous indicators, which were developed especially for this undertaking, consist of a transmitter located and operated by the machine in the lock wall, and a receiver operating an indicator at the control board. Both transmitter and receiver have a stationary and a rotating part. The stators have three-phase windings with leads brought from the transmitter and connected to three corresponding equidistant points of the receiver. These stator windings are not connected to a source of power, but are energized by induction from the rotors. The latter are bipolar and are connected in multiple and energized from a 110-volt, 25-cycle, single-phase source.

Movement of the lock machinery, with the connected transmitter rotor, induces voltage in the transmitter stator coils. This induced current is transmitted to the receiver-stator coils and duplicates in them. The rotor of the receiver, being energized by the external source in the same direction as that of the transmitter, is reacted upon by the polarized receiver stator until the magnetic axes coincide and the rotors of both transmitter and receiver are in the same relative position. Any difference

in the position of the transmitter and receiver rotors causes a difference of potential between the stator windings with a consequent flow of current and resultant torque, which again moves the receiver rotor to the same

In the case of the miter gates, the vertical operating shaft is connected to a shaft which operates the transmitter machine. The latter shaft is threaded and carries a nut on which is mounted a rack. The rack engages

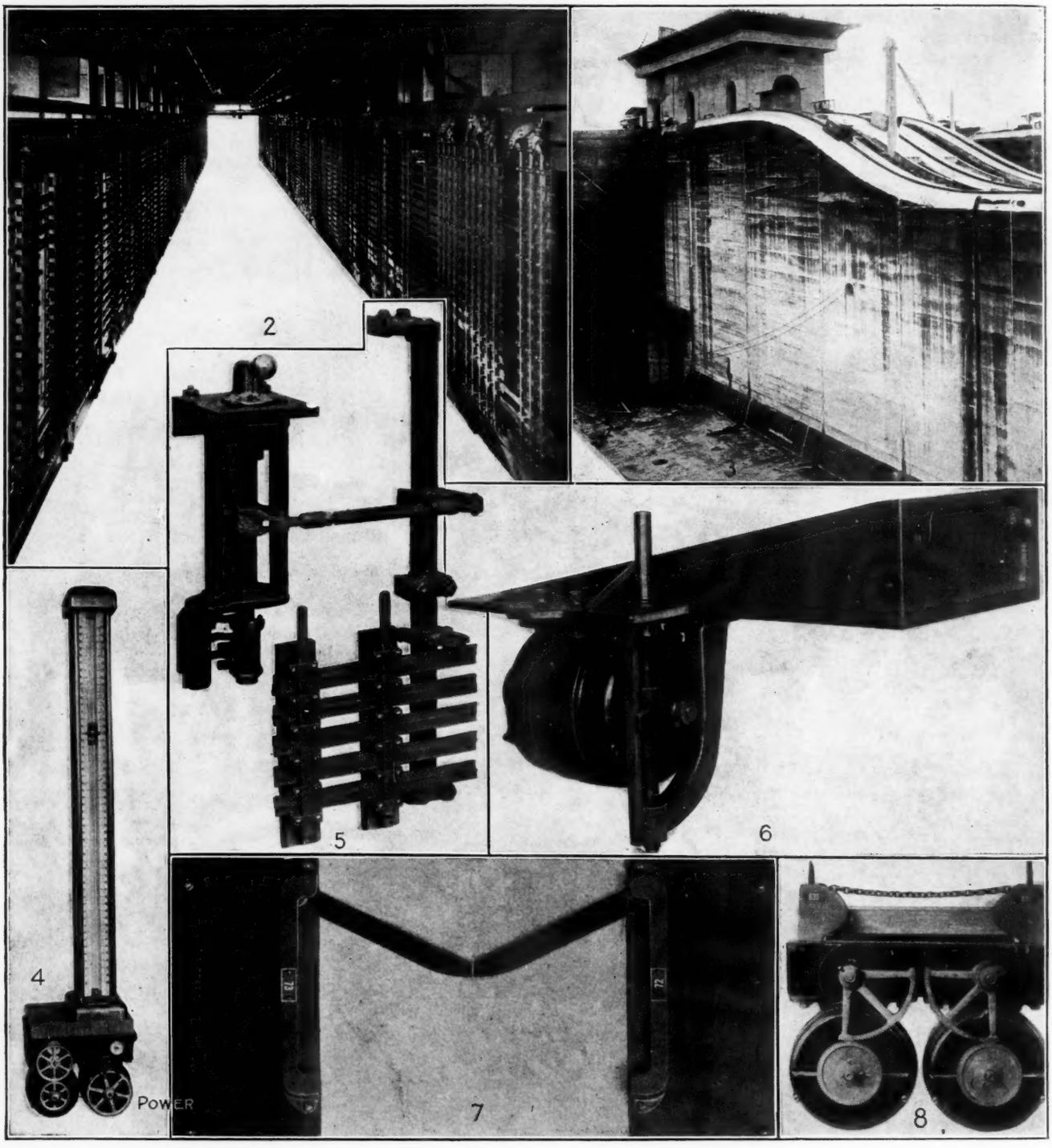


FIG. 2. INTERLOCKING SYSTEM UNDER CONTROL BOARD. FIG. 3. CONTROL HOUSE ON CENTER WALL OF UPPER LOCKS. FIG. 4. WATER-LEVEL INDICATOR. FIG. 5. CONTROL SWITCH AND INTERLOCK MECHANISM. FIG. 6. MITER-GATE TRANSMITTER. FIG. 7. MITER-GATE INDEX. FIG. 8. CHAIN-FENDER INDEX

relative position as that of the transmitter rotor. The receiver rotor follows closely and smoothly the movement of the transmitter rotor, and consequently imparts to the position indicator a movement identical with the movement of the lock machine, although on a scale reduced to the requirements of the control board.

a gear on the rotor shaft (Fig. 6), and this turns the rotor as the gates operate. The miter-gate indicator (Fig. 7) comprises a pair of aluminum leaves, shaped to correspond to the plan view of the top of the gate, which travels horizontally just above the top of the board, the hinge ends being connected to shafts extending down

through the surface of the board where they are geared to the receivers by bevel gears.

For the chain fender, the position-indicator transmitter is driven by the shaft which operates the limit switch that controls the stroke of the piston. The indication on the board is given by a small aluminum chain (Fig. 8) which, like the large chain, is raised and lowered, each end operating independently.

As the rising-stem valves occur in pairs, their position-indicator machines occur in pairs also. The transmitter rotor is driven by a shaft and gearing similar to that described for the miter gates. Each indicator is similar to a small elevator, a car being used to indicate the position of the valve gate. Both front and back of the shaft are fitted with opal glass marked with black lines for the ¼, ½ and ¾ positions. A small aluminum cage moves up and down in each compartment.

Water-Level Indicators

The specifications covering the water-level indication required an accuracy of $^1/_{20}$ of a foot in actual water level. In the transmitters and receivers for the machines described previously, the rotors turn less than 180 deg. with an inherent lag of 1½ per cent. between the transmitter and receiver rotors in this distance, which obviously prevents this arrangement from being employed to give the water-level indication.

It was found that if the rotors were revolved ten complete revolutions, the required accuracy could be obtained; but since this makes it possible for the rotors to be in synchronism every 180 deg., or in twenty different positions for the entire travel, the indicators would not indicate correctly if for some reason the transmitter rotors were turned more than one-half revolution with the power off. Therefore, the required accuracy was obtained by two sets of transmitters and receivers, one set connected to a fine index in which the rotors make ten complete revolutions and the other set connected to a coarse index operation, less than 180 deg.

The fine index is a hollow cylinder carrying a pointer, the length of the cylinder being such that when an aluminum ball representing the coarse index, which can be depended upon for coarse indication, is within the limits of the cylinder, the reading of the fine index is correct within the limits specified. The scales are illuminated by lamps in both base and top caps of the indicator.

For water-level indication, walls 36 in. square in the lock walls with communication to the lock by a small opening at the bottom of the wall to dampen surges contain a welded-steel box float, 30 in. square by 9 in. deep. A nonslipping phosphor-bronze belt transmits the movement of the float to a sheave fitted with pins on the transmitter mechanism, the pins registering with holes punched in the belt.

Mechanical Interlocking System

To make it necessary for the operator to maneuver the control-switch handles always in a certain order, corresponding to a predetermined sequence of operation of the lock machinery, and to prevent the operator in control of one channel from interfering with the machinery under the jurisdiction of the operator controlling the other channel, these control switches are provided with interlocks, as shown in Figs. 2 and 5. These are in two vertical racks under each edge of the board and some distance below, so that they may be easily inspected and oiled.

Vertical shafts operated by connecting rods from the control-switch shafts extend downward past the electrical parts for the operation of the interlocks. The interlock system is essentially a bell-crank mechanism, connecting the shaft of the control switch through a movable horizontal bar to a vertical operating shaft, which can or cannot move, according to the relative positions of the interlocking bars and dogs.

Interlocks prevent the chain fender from being lowered until adjacent miter gates have been opened, and also prevent the gates being opened until the chain is in the raised position. After the miter gates are closed, a miter forcing machine is operated by a control handle and locks the ends of the gates. This machine cannot be operated until the gates are closed.

Also the rising-stem valves of the side wall, next above or below a miter gate, must be closed while the miter forcing machine is open. As the miter forcing machine cannot be closed until the gates are closed, this means that the valves either above or below the gate must remain closed until the gate itself is closed, thus preventing the operator from creating a current of water around the gates while they are open, or being moved in opening or closing.

Either pair of rising-stem valves may be opened first, the interlock becoming effective when the first valve of the second pair of duplicates is opened. The cylindrical valves are interlocked so that if those on one side are opened, the ones on the other side will lock the opposite ten. This prevents careless cross filling between locks, which operation might be combined with the regular method and produce flooding. However, there may be times when it is desirable to employ cross filling to economize in the use of water from Gatun Lake in the dry season. For this reason this interlock is made removable by the use of a Yale lock, the key being retained by the chief operator.

In most cases the locks are divided into two unequal parts by the intermediate miter gates. This makes it necessary to divide the ten cylindrical valves into two groups of seven and three, respectively, for the long and short lengths. A selecting lever is provided for these interlocks and may be set as indicated by a nameplate on the lever to "three," "seven" or "ten," respectively; whereupon the corresponding valves are subject to that interlock, and the others of the group of ten are locked closed if three or seven only are used.

To withstand the humid atmosphere of the Isthmus, every insulated part, such as solenoid, relay, circuit-breaker and other coils, was impregnated with nonhygroscopic compounds. All small parts were made either of brass, copper, Monel metal, bronze or of sherardized iron or steel. Mica and treated asbestos lumber were used largely in place of fiber or wood.

Nearly two thousand special drawings were required in the fabrication, and in the construction of the boards there was employed more than 2½ miles of interlocking rod, about six million feet of control leads, 732 indicator motors, and 464 control switches.

All the lock-machinery motors, control, panels, centralized control boards, power-station generating apparatus, switchboards, transmission-line substation equipment for the wharf terminal cranes, etc., were manufactured by the General Electric Co.

Gas Power Department

A Unique Kerosene Carburetor

By J. A. Lucas

It is generally conceded by those who have acquired experience in the kerosene-carburetor field that the application of heat at some period of the travel of the fuel from the tank to the motor is necessary. The application of heat, however, must be carried out with extreme delicacy for if there is insufficient heat the process of vaporization is but slightly furthered, while if there is too much heat a chemical action known as "cracking" takes place, the kerosene splits up into its components, and the result is poor combustion, an odoriferous exhaust and an abundant deposit of carbon in the combustion chambers and other parts.

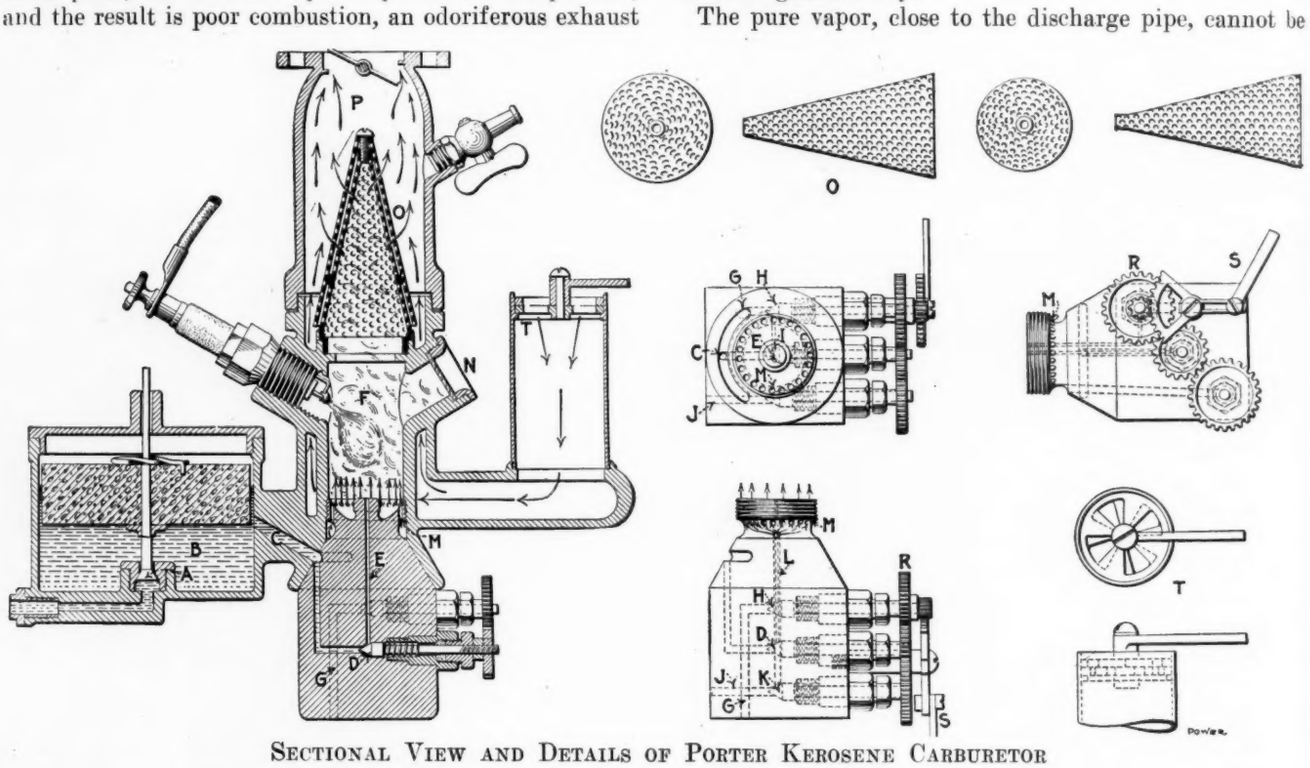
To those familiar with the more conventional lines of endeavor that have been followed in many attempts to solve the kerosene-carburetion problem the idea of applying heat by starting a fire, so to speak, in the mixing chamber of the carburetor itself, doubtless appears revolutionary. Yet it is by this very process that Alexander E. Porter, of 171 Broadway, New York, has succeeded in producing a kerosene vapor that differs widely from vapors formed by more familiar processes.

Preliminary Experiments

Before describing the construction of the apparatus it might be well to mention some of the preliminary results of its peculiar action.

While the carburetor is in actual service it is operated in the usual way by the suction of the motor. The experimental carburetor employed on the bench to demonstrate the principles of operation, however, is worked by compressed air, the office of which is to atomize the kerosene as the initial step in the process. The atomized kerosene (without the application of heat) issues in a spray which will rapidly condense on the hand and drip off. The instant a flame is applied, however, under the proper conditions of mixture, the characteristics of the vapor are entirely changed, and it becomes a thick, white vapor, strongly resembling steam, and the most interesting thing about it is that it is decidedly difficult to condense. It floats in the air like smoke and disappears in the same gradual way.

The pure vapor, close to the discharge pipe, cannot be ignited with a naked flame. Further out, however, where there is an admixture of air, it can be ignited with a match, and burns with a reddish flame. It is notable, however, that the flame gives off no smoke and produces no deposit.



Sectional View and Details of Porter Kerosene Carburetor

The same procedure in the case of the raw spray of kerosene results in the usual smoke and shower of small particles of black carbon in the air. The vapor is easily caught in a wide-mouthed bottle held over the discharge pipe of the carburetor, and, despite the coldness of the glass walls, remains visible in the bottle for 14 min. by the watch.

If a bottle of the white vapor is opened for a moment and a little air admitted, a match applied to the mouth will ignite the mixture, which will burn very rapidly and with a slight puff of flame, but without any deposit of carbon.

The conclusion naturally drawn from these demonstrations is that by this method it is possible to produce

a kerosene vapor that is sufficiently stable to withstand condensation considerably longer than is necessary to get it into the cylinders of a cold engine and ignite it. Furthermore, it would appear that under normal conditions combustion should take place without smoke or odor and without leaving deposits in the cylinders.

Referring to the illustration, showing a section through the carburetor and various details, the kerosene enters through a float valve *A* into the chamber *B*, thence through the passage *C* to the needle valve *D* and passage *E* to the atomizing chamber *F*. Atomizing air is drawn in by suction through the passage *G*, past the needle valve *H* and up through *I*, where being directed against the jet of kerosene, breaks it up into a spray. Air is also drawn in through the passage *J* past the needle valve *K* and up through passage *L*, thence through the annular holes *M*, as indicated by the arrows, and into the atomizing chamber *F*.

The spark plug ignites a small portion of this atomized kerosene mixture which burns as a small ball of fire and may be seen through the mica window *N*. The screen *O* prevents any flame from extending to the upper part of the carburetor. A single screen was first tried, but this was found to heat up after a few minutes and ignite the vapor in the upper chamber. The double screen, however, prevents this.

Because of insufficient air in the chamber *F* only a very small portion of the kerosene burns and furnishes just enough heat to break up the remainder into the form of a dense white vapor as previously mentioned. The proportions of air and kerosene entering this chamber are controlled by opening the needle valves, *D*, *H* and *K*, which are connected through gearing *R* (maintaining a constant ratio), the movement being controlled by a lever *S*, which operates in unison with the throttle.

The auxiliary air for supplying the proper mixture to the engine cylinders enters through the sliding throttle *T*, and up into the mixing chamber *P* as indicated by the arrows.

The spark plug is operated from an independent spark coil with a switch on the dash, and after the carburetor has been in operation a few minutes the current may be turned off and the ball of fire will continue to burn in chamber *F* and the carburetor will work as usual for a long period.

The motor starts, when cold, on the kerosene, apparently as easily as on gasoline. There is this difference between gasoline and kerosene, however:

Judging from the running of the motor, the kerosene burns more slowly than gasoline, there being apparently more gradual expansion and less of the explosion effect. The correctness of this would seem to be borne out, in a measure at least, by the fact that the motor will throttle down to a lower speed than with gasoline and will "hang on" when climbing a grade on high gear in a way that is rather surprising. While the speed of combustion is quite sufficient for all ordinary purposes, when a motor is raced to an excessive speed the power falls off sooner than with gasoline.

No trouble has been experienced from deposits in the cylinders or on the spark plugs except when the flame in the carburetor has been momentarily extinguished and raw kerosene vapor has been drawn in.

The vapor at the point of combustion is too rich to burn rapidly, burning beyond the screen is impossible and a backfire from the motor would have a tendency to extinguish the flame momentarily. Hence, there is no danger from the flame.

Most of the road test work with this kerosene carburetor has been with a large and heavy six-cylinder car, which is notoriously wasteful in the matter of fuel consumption, and the manufacture of which has been discontinued. The motor is rated at 60 hp. Running on gasoline the best average that could be made was five miles to the gallon of fuel. On kerosene with this carburetor the average rose to 10.5 miles per gallon, and the motor showed an appreciable increase in power, especially on hills.

Estimating Cost of Power in Producer-Gas Plant

By G. A. Field

The following is a preliminary estimate of the costs of a 300-hp. gas engine and producer. The engine is assumed to be a three-cylinder, vertical type and the producer to be of the bituminous water-sealed type with superimposed vaporizer. The estimate includes the first cost of the engine and producer erected on foundations, together with all piping and gas-cleaning apparatus and coal-handling machinery, also the annual cost of operation.

It is understood that the factors and assumptions used will vary through a wide range, depending on the following:

1. The nature of the service required, whether to drive electric generators, air compressors, ordinary mill machinery, etc. In many of these cases it might be advisable to plot the load curve before deciding upon the size of the units.

2. The quality or grade of bituminous coal to be used. This has a marked influence upon the quality and class of labor required, the efficiency of the producer, the cost of the gas-cleaning apparatus, the cost of disposing of the tar, etc.

3. The charges in first cost to freight, and incidental expenses will obviously depend largely upon the location, shipping facilities, etc.

4. The load factor. The fuel consumption per brake horsepower-hour increases as the load decreases from full load. The other operating costs per brake horsepower also increase, though in different ratios.

As a result of these variable conditions each case should be treated in a more or less special manner to give the best satisfaction to the parties concerned.

The estimate does not include the cost of the water required for the plant. This may be easily determined for the particular case by allowing 10 to 13 gal. per b.hp.-hr. for the engine and 3 to 4 gal. per b.hp.-hr. for the producer.

The cost of the building and land is also omitted.

Determination of Size of Producer

The B.t.u. in the gas required per brake horsepower-hour from full load to three-quarter load is assumed to be 12,000.

$$300\ hp. \times 12{,}000 = 3{,}600{,}000\ B.t.u.$$

required in the gas per hour. Assume the producer efficiency to be 65 per cent.

$$3{,}600{,}000 \div 65\ per\ cent. = 5{,}538{,}461\ B.t.u.$$

in coal per hour. Assume the heat value of coal to be 13,500 B.t.u. per lb.

$$5{,}538{,}461 \div 13{,}500 = 410.25 \ lb.$$

of coal per hour. Assume a gasification of 10 lb. of coal per square foot of grate area per hour

$$410.25 \div 10 = 41.02 \ sq.ft.$$

of grate area. An internal diameter of 7 ft. 3 in. would satisfy this requirement.

Alternate method: As in the above case the B.t.u. required in the gas per hour equals 3,600,000. Assume gas at 135 B.t.u. per cu.ft. (low value).

$$3{,}600{,}000 \div 135 = 26{,}666.6 \ cu.ft.$$

of gas per hour. Assume 65 cu.ft. of gas generated per lb. of coal per hour.

$$26{,}666.6 \div 65 = 410.25 \ lb.$$

coal per hour. Assume a gasification of 10 lb. of coal per sq.ft. of grate area per hour

$$410.25 \div 10 = 41.02 \ sq.ft.$$

of grate area, which is satisfied by an internal diameter of 7 ft. 3 in. This represents a gasification of 1.36 lb. per b.hp.-hr.

FIRST COST OF PRODUCER AND ENGINE

Producer @ $2.25 per hp.	$675.00
Producer lining @ 0.25 per hp.	75.00
Labor for lining @ 0.10 per hp.	30.00
Blower for starting @ 0.10 per hp.	30.00
Centrifugal scrubber @ 1.50 per hp.	450.00
Intermediate scrubber @ 0.40 per hp.	120.00
Tower scrubber @ 0.60 per hp.	180.00
Piping and auxiliaries (blast) @ 0.50 per hp.	150.00
Moisture separator @ 0.20 per hp.	60.00
Water and gas piping @ 0.60 per hp.	180.00
Filling for scrubbers @ 0.25 per hp.	75.00
Erection @ 1.25 per hp.	375.00
Total	$2400.00
Coal handling apparatus @ $6.00 per hp.	$1800.00
Total	$4200.00
Add for starting up, auxiliaries, freight, travel, manufacturer's profit, etc., about 30 per cent. of $4200.00	1260.00
Total first cost of producer and equipment	$5460.00
Cost of engine on foundation	$13,000.00
Total first cost of engine and producer	$18,460.00
First cost of engine and producer per hp.	$61.53

OPERATING COST OF PRODUCER

Coal gasified per hr. =	410.25 lb.
Standby loss, 2 per cent. of 410.25 =	8.20 lb.
From these figures any variation in loss in fuel due to shut-down of engine may be calculated.	
Assume a special case calling for service 10 hr. a day for 313 days of the year; overhauling, etc., to be done Sundays and at night.	
313 days @ 10 hrs. a day =	3130 hr.
3130 × 410.25 = 1,284,082 lb. of coal converted into power per year.	
Total number of hr. per year = 8760	
8760 — 3130 = 5630 standby hr.	
5630 × 8.2 = 46,166 lb. standby cost per year.	
Total consumption per year in tons of 2000 lb. = 665.12.	
Coal—665.12 tons @ $2.00 per ton =	$1330.24
Repairs and maintenance @ 1.75 per cent. of 5460.00 =	95.60
Interest, depreciation, taxes, insurance @ 13 per cent. =	709.80
Supplies, about	30.00
Labor @ 4.00 per day (365 days)	1460.00
Total annual cost of producer =	$3625.64

OPERATING COST OF ENGINE

300 b.hp. for 3130 hr. = 939,000 b.hp.-hr.	
First cost of engine erected on foundation =	$13,000.00
Depreciation @ 10% of 13,000.00 =	$1300.00
Supplies @ 0.03c. per b.hp.-hr. =	281.70
Lubricants @ 0.04c. per b.hp.-hr. =	375.60
Repairs and maintenance @ 0.15c. per b.hp.-hr.	1408.50
Interest @ 5 per cent.	650.00
Annual cost of engine =	4015.80
Annual cost of producer =	3625.64
Total	7641.44
Power required to operate centrifugal scrubber is about 5 per cent. of total output.	
Cost per year for operating scrubber =	382.00
Total annual operating cost of plant =	$8023.44
Total annual cost per b.hp.	$26.74
Operating cost per b.hp.-hr.	0.85c.

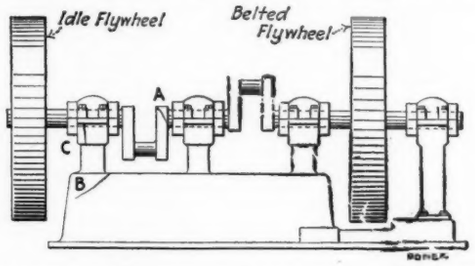
[The low cost per brake horsepower-hour of 0.85c. is due largely to the high load factor assumed during the hours of regular operation. The same method of figuring the results is applicable to other load conditions.—EDITOR.]

CORRESPONDENCE

Broken Crankshaft

Referring to the broken crankshaft described by M. W. Utz, in the Dec. 9 issue of POWER, and shown again herewith, the fact that the shaft broke at point *A* does not prove that the break was caused by an excessive torsional stress, but the writer is inclined to think that such stress combined with the possible disalignment may have contributed considerably to the failure.

The writer has never favored gas-engine designs of the type shown, where heavy flywheels are placed on both sides of the crank. If two flywheels must be used for the sake of bettering the balance, there is no good reason why the idle flywheel should not be made considerably lighter than the belted one. Where the idle wheel is just as heavy, it stores just as much energy as the other, and during the cycle in which it gives up its energy to the driven



CRANKSHAFT, SHOWING LOCATION OF FRACTURES

machine or machines that energy must be transmitted through a long, crooked shaft, as clearly shown in Mr. Utz's drawing. Where a load is suddenly applied, or in sudden binding or catching of the driven machines, as often occurs, there is a possibility of crankshaft failure, even though the load which is being pulled is light normally.

Where a single flywheel is used, or where the idle wheel is small, this danger is practically eliminated. True, a gas engine looks better with two equal sized flywheels, even though the load which is being pulled is light normally

New York City. S. F. WILSON.

From the letter on "A Broken Crankshaft," printed in the issue of Dec. 9, page 825, it appears likely that synchronous vibration played a part in the accident. There would be two critical speeds for the engine, one affecting the torsional moment and the other the bending moment. The sketch looks as if the latter were the trouble. As a check to this it would be interesting to know if there were any signs of torsional fracture at *A*. The fact that the engine broke when carrying no load is strong evidence in support of synchronous vibration.

The engine was probably running below the critical speed normally. Some of the babbitt being wiped out of the bearing would weaken the directional restraining power of this bearing and consequently increase the period of natural vibration of the shaft or reduce the critical speed, making it come down to that of the engine speed or dangerously near to it.

JOHN AIREY.

Ann Arbor, Mich.

Refrigeration Department

Testing Ammonia Piping

By Gilbert Crawford, Jr.

Recently the ammonia pipe coils in a new freezing tank were in place before the compressor was entirely erected. It was desired to test these coils immediately so that the framework could be placed on the tank. As there was no air compressor available, the erectors had to find some method of getting a test pressure. Someone suggested

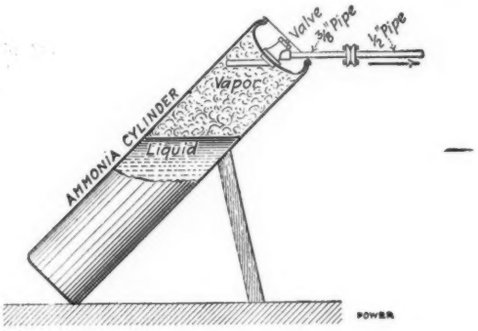


Fig. 1. Position of Ammonia Drum

that a little ammonia be injected into each coil in turn, and a test for leaks made with the ammonia. This suggestion was taken as a starting point and a plan was developed for the easy and economical testing of each coil with ammonia vapor pressure.

Instead of attaching a cylinder of ammonia to each coil, a connection was made between the expansion valve of the first coil and the vapor space of the cylinder. In this way vapor could be passed into the first coil at a pressure corresponding to the temperature of the liquid ammonia in the cylinder. The cylinder was so placed that no liquid could pass out through its valve. In this way the minimum of ammonia was used to produce the pressure.

Fig. 1 shows how the cylinder was placed so that only vapor could be drawn therefrom. In the position shown, it was easy to make the necessary pipe connections to the first coil through the expansion valve X_1. Fig. 2 shows the general layout of the coils. The liquid header was not used for the test, and is not shown. The expansion valves, with the exception of that connected to coil 1, were used as blowoff valves, for allowing the air to be expelled from each coil. The expansion valves are represented by X_1, X_2, X_3, etc., while G_1, G_2, G_3, etc., represent the gas-return valves. The main gas-return valve is shown at G.

In making the test all the valves were closed except G_1, X_1 and G which was "cracked" a little. By opening the small valve at the cylinder, ammonia vapor passed into the first coil, driving the air before it out through G_1 and G. When ammonia was detected escaping at G, this valve was closed. The vapor from the cylinder continued to pass into coil 1 until the pressures were equalized. This condition was known to exist when ebullition ceased in the cylinder. Coil 1 was then carefully inspected for leaks, every joint being examined.

When it was certain that there were no leaks in coil 1 valve X_2 was opened slightly, then G_2. This allowed the pressure from coil 1 to drive the air from coil 2, after which X_2 was closed. After the pressure in coils 1 and 2 and the cylinder equalized, coil 2 was inspected for leaks.

For testing the next coil, a slightly different procedure was followed. Valve X_1 was closed, X_3 was opened a little, then G_3. Part of the gas from the first two coils passed into the third coil, driving the air ahead of it out through X_3, after which X_3 was closed. The gas pressures in the three coils now equalized at a pressure less than that in the cylinder. Coil 2 was cut off from the others by closing G_2. Then X_1 was opened and the pressure in the cylinder equalized with coils 1 and 3. Coil 3 was then inspected for leaks.

Coil 2 now contained gas at a pressure lower than that existing in the cylinder. By closing all valves, then allowing coils 2 and 4 to equalize, then shutting off coil 2 and opening up coil 3 until coils 3 and 4 were equalized, then shutting off coil 3, it was possible to fill coil 4 with gas at a pressure nearly equal to the cylinder pressure. The final increment of gas was supplied from the cylinder.

This method was followed with the remaining coils of the tank, so that at the end, the last and the first coils

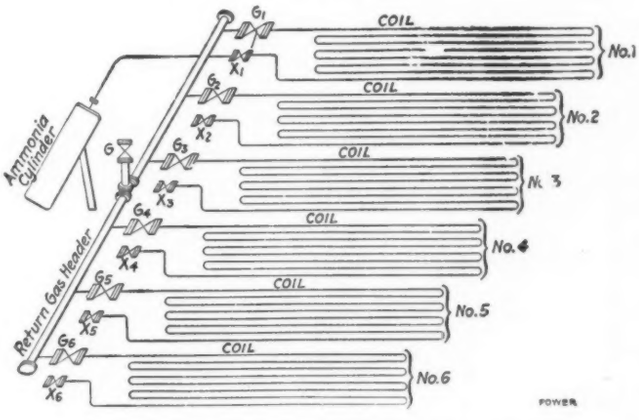


Fig. 2. Layout of Coils to Be Tested

were filled with gas at the pressure existing in the cylinder, while the other coils contained gas at various lower pressures.

As the liquid in the cylinder was at a temperature of 65 or 70 deg. F. at the beginning of the test, and as it did not cool much, a pressure in the coils was obtained of from 103 to 114 lb. per sq.in. gage. The coils in the tank amounted to 4800 ft. of 1¼-in. pipe, having an internal volume of about 50 cu.ft. This volume of gas at the above mentioned pressure, saturated, would weigh about 21 lb. As the gas was considerably superheated, and as all the coils were not filled with high-pressure gas at the end, no such quantity of ammonia was used.

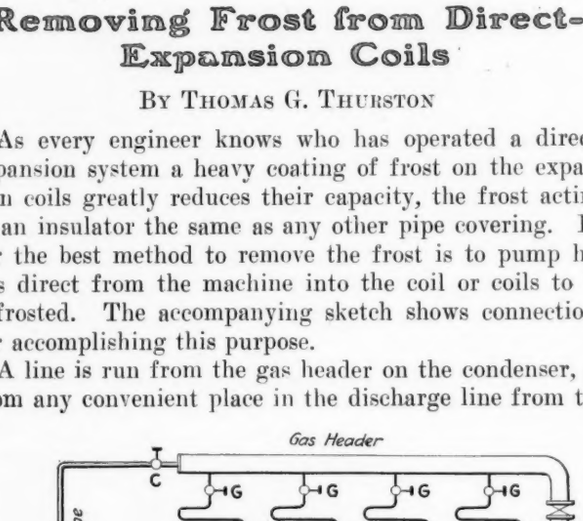
Later on to make sure that this test was complete an air and ammonia pressure of 250 lb. per sq.in. was applied to the coils, and not a leak discovered. The cost of this test in time and ammonia was small, and the results were satisfactory.

Removing Frost from Direct-Expansion Coils

By Thomas G. Thurston

As every engineer knows who has operated a direct-expansion system a heavy coating of frost on the expansion coils greatly reduces their capacity, the frost acting as an insulator the same as any other pipe covering. By far the best method to remove the frost is to pump hot gas direct from the machine into the coil or coils to be defrosted. The accompanying sketch shows connections for accomplishing this purpose.

A line is run from the gas header on the condenser, or from any convenient place in the discharge line from the



Sketch of Layout

compressor, to the liquid line. To remove the frost from a coil close valve *A* in the liquid line and pump the expansion coils and liquid line down to zero. Shut off the water from the condenser, open valves *B* and *C* in the hot-vapor line and let the hot gas blow through the liquid line until the liquid is blown out. This will be indicated by the frost disappearing on the low-pressure side of the expansion valves. Then close the expansion valves *D* on the coils which do not need defrosting, and on the coils from which the frost is to be removed close the suction valves *E* until they are barely cracked off the seat. Open the expansion valves wide on these coils and run the machine slowly. The frost will loosen and come off in a short time.

During these operations the condenser pressure must be watched as in a short time it is liable to run up high enough to blow the relief valves. If the condenser pressure runs up to the limit before the liquid line and coils are pumped down to zero, let it go at that, as it is not absolutely necessary to pump down to zero pressure. If the condenser pressure becomes too high while pumping hot vapor, turn sufficient water on the condenser to keep it down.

After the coils have been defrosted turn the water on the condenser, close the valves in the hot-vapor connection, close the expansion valves on the defrosted coils, open them on the other coils and open valve *A* in the liquid line at the receiver. After the machine is running normally open gradually the suction valves *E* on the defrosted coils, as considerable liquid accumulates in these coils during the cleaning process. If the valves are opened wide at once the liquid will be carried back to the machine before it evaporates and freeze the machine up if nothing worse.

Another and safer way to proceed after the coils are defrosted is to turn the water on the condenser, shut the valves on the hot-gas connection and open the expansion valves on the coils not cleaned. Then operate the machine and draw the liquid accumulated on the defrosted coils through the liquid line into the other coils until frost shows again on the cleaned coils. Then set the expansion valves on the latter coils for regular running, open their suction valves and the main liquid valve. This way of removing the liquid from the defrosted coils is safer than the method just described.

In case there is no hot gas connection, the hot gas can be pumped through one of the condenser coils and the liquid receiver by closing the liquid valves *F* between the condenser coils and liquid header on all the coils but one. Close the gas valve *G* on this coil and pump the coil, liquid header, receiver, liquid line and expansion coils down to zero if the head pressure will permit. The condenser coil which is used for this purpose should have the water drained from it in case the condenser is of the double-pipe type; otherwise it is liable to freeze while pumping out, or when pumping hot gas the water will condense some of the gas. After the system has been pumped out, open the gas valve *G* and proceed as before.

On a large number of modern compressors a separate pump-out line is run from the suction side of the machine to the condenser, liquid receiver and liquid line. In this case the pump-out line can be used for a hot-gas line. The section of the line to the compressor is shut off and a couple of valves opened on the pump-out line on the condenser coils and the pump-out connection to the liquid line. This allows the hot gas to flow through the condenser pump-out valves and the section of the pump-out line between these and the liquid line pump-out valve into the liquid line. This holds good only where the pump-out connection is at the top of the condenser. If it is at the bottom one of the coils will have to be pumped out and the hot gas pumped through it into the pump-out connection.

Alaska's Coal Development Retarded—The development of the coal resources of Alaska, according to the United States Geological Survey, has been held back on account of the lack of legislation permitting the exploitation of the known extensive and valuable fields. In 1912 the commercial product of coal in Alaska amounted to only 355 tons.

Editorials

Progress in 1913

Large is the word best suited to point out the tendencies of the year. In nearly every branch of the field units have been installed which are greater in capacity or size than ever before, and from all indications it would appear that present achievements have only whetted the appetite of manufacturer or user for a continuation in the same direction. The wonder of yesterday is only commonplace today and the future promises marvels up to the physical or efficient limits.

Early in the year the Inverness Railway & Coal Co. of Inverness, Cape Breton Island, ordered a Nordberg Corliss duplex double-drum hoisting engine, 34 and 34 by 72 in., designed to pull a 41,000-lb. load up a 10,000-ft. incline of 16 deg. at the surface and 35 deg. at the bottom. From the standpoint of rope stress and length of cable the hoist is thought to be the largest ever built, although a number of hoists with larger cylinders have been installed.

In Pittsburgh, a De Laval centrifugal pump having a rated capacity of 100,000,000 gal. per 24 hours against a total head of 56 ft. was ordered for the Ross pumping station. This is reported to be the largest steam-turbine-driven centrifugal pump in this country. Two immense Bethlehem Steel Co. pumping engines of the vertical triple-expansion high-duty type have been put in service at the new Mission Street station. Each pump has a capacity of 7,000,000 gal. per 24 hours, against a head of 495 ft.

The Mesta Machine Co. has built for the Gary works of the United States Steel Corporation, a 44 and 76 by 60-in. twin-tandem-compound geared reversing engine weighing 1000 tons and capable of developing the enormous output of 25,000 hp.

In boilers, those at the Delray station of the Detroit Edison Co. still hold the record for output and square feet of surface. Nominally they are rated at 2350 boiler horsepower on a basis of 10 sq.ft. of heating surface. It has been found possible to carry for an hour on a single unit, a load of 11,000 kw. In the Conners Creek station to be installed on Belle Isle, it is planned to use the same type of boiler with equal heating surface, but with the furnace modified so that it will be possible by forcing to continuously carry the enormous load of 13,300 kw. on one boiler. During the year a number of large boilers have been installed but none of them has over half the heating surface or steaming capacity of those at the Delray station.

Other large units of their particular type are a 5000-hp. gas engine, a 25,000-kw. alternating-current generator, a 3750-kw. direct-current machine, a Le Blanc jet condenser capable of condensing 150,000 lb. of steam per hour and a 10,000-hp. turbine at Keokuk, weighing 1,600,000 lb. The magnitude of the unit last named is measured in weight rather than in capacity as there are several hydraulic units developing twice the power.

Steam Turbines

It is in the steam-turbine field that the most rapid development has been made in the capacity of a single unit. Not long ago a 20,000-kw. unit was considered the last word in turbine design. During the year a unit of this size and one having a capacity greater by 5000 kw., have been installed and advance reports state that orders have been placed for four 30,000-kw. units and one having a capacity of 35,000 kw. In these larger sizes it is of interest to note that all are horizontal machines, even at the Fisk Street station of the Commonwealth Edison Co., of Chicago, where all of the previous units, aggregating 120,000 kw., are of the Curtis vertical type.

Naturally these giant units call for large auxiliaries. The condenser for the 25,000-kw. turbine mentioned above has 39,300 sq.ft. of tube surface which will condense 300,000 lb. of steam per hour, and the condenser for the 20,000-kw. unit has 35,000 sq.ft. With the former unit the usual air pump has been replaced by the kinetic air-ejector for the first time in this country. A small gain in vacuum means a great deal on so large a turbine and the operation of the new apparatus will be watched with interest. Temporarily the generator is the largest alternating-current machine in the field, and in connection with the plant there is an open feed-water heater having a capacity to heat 300,000 lb. of water per hour from a temperature of 65 to 130 deg.

Before passing from this station it may be of interest to state that a steam rate of 11.25 lb. per kw.-hr. at the normal rating of 20,000 kw. is expected of the larger unit. This betters by a little over one-half pound the remarkable rate obtained some two years ago on a 6000-kw Parsons turbine in the Dunston power station of the Newcastle-on-Tyne Electrical Supply Co.

In the smaller turbines the Terry Steam Turbine Co. has developed a multistage machine of the return flow type and the De Laval Co., a velocity stage turbine consisting of two or more regular De Laval wheels with a row of reversing blades between each pair. The mixed-pressure turbine has been given considerable attention. With its two sets of nozzles it will run with equal efficiency on high- or low-pressure steam and can be operated independently of the unit on which it depends for its supply of exhaust steam. In this feature it is an improvement over the usual exhaust-steam turbine.

During the last few months Europe has not been standing still. Ferranti in England has built a turbine of 3000 kw. capacity which runs on 9.4 lb. of steam per kw.-hr., or about 7.5 lb. per i.hp.-hr. LeBlanc in France is advocating a return to single-stage turbines running at very high velocities made possible by the new high-tensile strength steels. Schmidt, the superheater man, is making experiments with very high pressures and temperatures that promise results for steam comparing favorably with those of the internal-combustion engine.

Engine Progress

In reciprocating engines there has also been advancement. Perfection of design and construction has resulted in some remarkably low steam rates and the use of superheated steam in an engine of the poppet-valve type has made possible a rate of 15.24 lb. per i.hp.-hr., noncondensing. The old reliable and efficient Corliss is still in the field and will be for years to come, but the tendency is away from this standby of two generations and toward the high-speed engine and some of the recent importations from Europe built on the Lentz and Unaflow principles. The year has also seen the first American locomobile which, on test, has developed an indicated horsepower-hour on 9.65 lb. of steam and 1.51 lb. of Holwick coal.

A number of rotary engines and so-called rotary air-pumps have "seen the light" and there has been a new design of triplex pump in which cams take the place of cranks in producing the reciprocating motion.

Electrical Advancement

Perhaps the most important new development during the past year in the electrical field, was the half-watt nitrogen-filled tungsten lamp brought out by the General Electric Co., as a result of several years work by Dr. Irving Langmuir. There has been little change in generators and motors except in the size of the former to keep pace with the increasing capacity of steam turbines. The largest alternating-current generator installed during 1913 was a 25,000-kw. machine at the Commonwealth Edison Co., of Chicago, although still larger ones have been ordered, and the largest direct-current machine, having a capacity of 3750 kw., was installed at the Canal Road plant of the Cleveland Electric Illuminating Co. For alternating-current transmission, 150,000 volts still remains the upper limit, work on the Big Creek Development in California, at which this is employed, having progressed steadily during the past year. In direct-current transmission, however, a bold step was taken in the decision to employ the Thury system at 90,000 volts to transmit 20,000 kw. from the Trollhattan Falls in Sweden to Copenhagen, Denmark.

Water Power

The two most notable hydroelectric developments which were formally put in operation during the past year in this country are the Mississippi River Company's plant at Keokuk, Iowa, and the Hales Bar Plant on the Tennessee River. The former is notable chiefly because it will ultimately be the largest single water-power development, aggregating about 300,000 hp., although the immediate plans call for only 150,000 hp. The Hales Bar plant, although designed for only 44,000 kw., possesses a number of unusual features and was completed only after eight years of labor, during which a chain of unforseen difficulties was encountered, making the initial cost greatly exceed that estimated.

Work on a number of smaller hydroelectric developments has been started and several of the existing systems have been extended. Among these may be mentioned the Appalachian development on the New River, Va., consisting of five plants, aggregating 75,000 hp., two being already completed; the New England Power Company's developments on the Deerfield River, four plants being completed and a fifth under construction; a 12,000-hp. plant on the Coon Creek Rapids near Minneapolis; the Turner's Falls development on the Connecticut River; and in the Pacific Coast states extensions to the systems of the Washington River Power Co., the San Joaquin Light & Power Co. and the Pacific Gas & Electric Co. The last named company now has sixteen plants, steam and water power, totaling about 300,000 hp. In Switzerland construction has been started on a 5312-ft. head plant which holds the record as the highest head thus far to be developed.

The Gas Engine

Among special applications in the gas-power field, Professor Hopkinson's method of cooling gas-engine cylinders by direct injection of water onto the piston and exhaust valves, has attracted widespread attention. Dr. Low, in England, is still experimenting with his internal-combustion engine into which coal is fed directly. In the original design, tubes containing coal extended across the explosion chamber, the gas being driven off from the coal by the heat surrounding the tubes and drawn into the chamber. Spiral conveyors extended through the tubes and by their rotation urged the coal forward. In the later design these conveyors, which lost their rigidity under the high temperature, are abandoned, and the pipes made larger and vertical so that the coal feeds forward by its own weight.

The installation of Humphrey gas pumps for the Metropolitan Water Board has been completed and is now in operation. Last summer also marked the installation of the 5000-hp. gas engine (the largest in this country) at the plant of the Ford Motor Co., Detroit. The first Diesel locomotive was also completed by Sulzer Bros. of Winterthur, Switzerland, and put in service on the Prussian-Hessian State Railway.

In last year's review we called attention to a number of semi-Diesel engines that were being advanced by American manufacturers. The past year has seen several well known American firms take up the building of heavy oil engines of the regular Diesel type.

Refrigeration

During the year the refrigerating field has been given a number of new tables on the properties of ammonia. Perhaps the best of these was the set issued by the Engineering Experiment Station of the University of Illinois. More accurate experimental work than has been conducted in the past may render necessary some revision of the tables, but within the limits of practical working they may be safely accepted as standard.

The Bureau of Standards at Washington has been making some interesting tests to determine the latent heat of ice, and have found that a value of 143.5 B.t.u. per lb., instead of the 144 B.t.u. now used in this country, is approximately correct. The bureau is to continue its good work. Apparatus is now ready for making tests on the resistivity of insulation, and there will also be some work on the properties of ammonia and perhaps some of the other refrigerants commonly used. Pennsylvania State College now has a large thermal testing plant and in the past year has been testing some of the standard forms of insulation. Accurate standards have been badly needed in the refrigerating field so the work now under way should prove of immense benefit. The Third International Congress of Refrigeration held re-

cently in Chicago afforded Americans the opportunity of a generation. Engineers from twenty different countries met and discussed topics of pressing interest and a more uniform and concentrated effort in advancing the art is to be expected as a direct result.

Marine Tendencies

On the high seas nothing of any great import has come to pass. The quadruple-screw turbine-driven Hamburg-American liner "Imperator," with 62,000 hp for driving her propellers and 203,000 sq.ft. of boiler-heating surface, was put into service. The 20,000-ton collier "Jupiter" of the United States Navy also went into commision. She is equipped with a high-speed Curtis turbine, the first of a type, and the reduction in speed for efficient propeller operation is accomplished electrically. The turbine drives a 5450-kw., two-pole generator which supplies current to two 2750-hp., 36-pole motors, so that the ratio of reduction is practically 1 to 18. Test data from this ship may possibly have some influence in reconverting navy officials to turbine propulsion.

Oil fuel is fast becoming a question of urgent importance to the navy. Last year 30,000,000 gal. were used and in the near future it is likely that this amount will be increased over fourfold. Within the last month Secretary Daniels has recommended that the navy secure its own oil wells and do its own refining. A constant supply at a low cost would then be assured and the exclusive use of oil fuel might be adopted with reasonable security.

Engineering Societies

Among the various engineering societies in this country the American Society of Mechanical Engineers has had a very successful year. The attendance is increasing, more papers of a high standard are being presented and the standing committees are doing excellent work. The old question of flanged fittings was up at the annual meeting. The committee had formulated a compromise with the manufacturers which would do away with the dual standard, but a vote before the society as a body resulted in laying the matter over until the next meeting. At the invitation of the Verein deutscher Ingenieure, 300 members and friends of the society attended the spring meeting of the German society at Leipzig. The visitors were taken all over Germany in a special train, their entertainment including fifteen formal banquets and receptions, fourteen lunches and collations, twelve concerts and lectures, and special performances and ten excursions by rail or boat.

Boiler and Flywheel Explosions

Boiler accidents and explosions in this country for the first six months of the year total 234, in which are included stationary, marine, locomotive, portable and heating boilers. Those killed number 68, and 234 were injured. Assuming that the last six months will duplicate these figures, the totals for the year will be 468, 136 and 468, respectively. Besides there were three economizer explosions, killing six and injuring twenty. There were also a number of steam-pipe ruptures, air- and elevator-tank explosions, cylinder-head breaks, etc., which in the Hartford Co.'s compilations are usually included under the broad head of "boiler explosions." With these items included, the records for 1912, in the same order as before, give 537, 278 and 392. That the number of heating-boiler explosions was particularly large would indicate that more attention should be given to the inspection of this type of boiler. The flywheel explosions totaled about fifty for the year, and from the records of the first eight months the death list should be about twenty, with serious injuries to thirty more.

License and Inspection Laws

Year after year the list of fatalities reaches this same high limit or perhaps exceeds it, and relief from this inexcusable slaughter cannot be expected until every state in the Union has stringent license and boiler-inspection laws which are intelligently enforced. There has been agitation in quite a number of the states, but the net results appear to be an engineer's license law for California, a boiler-inspection law for Indiana, revised laws for Montana, a license law for New Jersey, and new boiler regulations for the province of Ontario. With the aid of the National License Committee of the National Association of Stationary Engineers other states will no doubt soon fall in line, and the recent adoption of standard boiler specifications by the National Association of Tubular Boiler Manufacturers should also help on the side of safety.

Blazing the Way

Appropriations for the fiscal year ending June 30, 1915, asked by the Bureau of Mines, indicate that it is to continue on an increased scale its valuable work relating to fuel and mines. Of this $135,000 is to be devoted to fuel testing, $10,000 for the equipment of a testing plant and $30,000 for investigating petroleum and natural gas.

At last it would appear that some attention is to be given to the achievements of science and industry. The land is full of libraries, there are museums of natural history and art, but there is nothing worthy of the name to record and exhibit the "milestones" in engineering development. Steps have recently been taken to found a group of museums in New York City, "The Museum of Peaceful Arts," which will consist of twenty buildings costing $30,000,000 and requiring two to three million per year for maintenance. It will be divided into branches on electricity, steam navigation, safety appliances, mechanical arts, mining, architecture and other subdivisions of equal interest and usefulness. The idea is splendid and if carried through to completion, will result in great good to the fields represented.

Isolated Plant vs. Central Station

Much in the same way as in previous years, the central station and isolated plant are carrying on their fight for supremacy. It is becoming evident to both sides that there is a field for both, and that the decision placing a plant in either must be based on accurate records and statistics fair to both sides. Probably the most comprehensive test ever conducted is that just completed by representatives of the City of New York and the Edison Co. The plant in the Hall of Records has been the scene of action, and for a full year a complete plant test has been under way. Complete operating data will be available for all seasons of the year, and upon these figures it will be decided whether the city is to continue to generate or buy its current. That municipal operation of a power plant is practical and that efficient results may

be obtained in such a plant has again been shown in Pasadena. The sixth annual report on this plant, published in our issue of Oct. 28, contains some interesting facts.

The Honor Roll

Among the men honored in the mechanical field were George Westinghouse, who was awarded the Grashof medal; Prof. H. K. Onnes, of Leiden, given the Nobel prize in physics, and Prof. A. Werner of Zurich, the prize in chemistry. The following men were elected as presidents of the various engineering societies in this country, relating to the field: James Hartness, president of the American Society of Mechanical Engineers; C. O. Mailloux, president of the American Institute of Electrical Engineers; Joseph B. McCall, president of the National Electric Light Association; James R. Coe, president of National Association of Stationary Engineers; John F. Hale, president of the American Society of Heating and Ventilating Engineers; Henry Torrance, Jr., president of the American Society of Refrigerating Engineers.

Necrology

Those of prominence who passed away during the year just closed were the following: Victor Hugo, manager and chief inspector of the St. Louis department of the Hartford Steam Boiler Inspection & Insurance Co.; John Fritz, famous in the steel industry; Dr. C. G. P. de Laval, inventor of the turbine and separator bearing his name; William Henry White, a professor of naval architecture and originator of the modern battleship; W. H. Fletcher, vice-president of the Consolidated Iron Works; Philip H. Diehl, founder of the Diehl Manufacturing Co.; Victor Dwelshauvers-Dery, a distinguished scientist noted for his researches into the properties of steam and its action in the engine cylinder, and his work for the advancement of engineering education; James Tangye, one of the Tangye brothers so well known in English mechanical circles; Samuel L. Moyer, first vice-president and general manager of the Lunkenheimer Co.; Francis M. Rites, inventor of the inertia governor bearing his name; Horatio A. Foster, author of the "Electrical Engineer's Pocket Book" and chief appraiser for J. G. White & Co.; Horace Greeley Burt, ex-president of the Union Pacific R.R. system and later chief engineer of the Chicago Association of Commerce Smoke Abatement Committee; Stephen D. Field, inventor of an electrical elevator and electrical devices relating to telegraphy; Bernard Schuchardt, head of the well known firm of Schuchardt & Schütte; Anthony N. Brady, president of the New York Edison Co. and a number of other corporations; Eugene O. Reuleaux, general superintendent of the American Rotary Valve Co.; Dr. Rudolf Diesel, famous through the invention of the engine bearing his name; William Cary Seldon, a consulting mechanical engineer who had much to do with the early development of steam navigation in the United States Navy; George C. Karme, secretary of the Hills-McCanna Co.; John F. Ensign, chief of the division of locomotive-boiler inspection of the Interstate Commerce Commission; Edwin T. Moore, secretary and treasurer of the Coatesville Boiler Works; Charles Tellier, familiarly known as the "father of refrigeration"; Edgar Penney, president of the American Society of Refrigerating Engineers in 1908; A. J. Frith, associate professor of mechanical engineering at Armour Institute, Chicago.

Concerning the "Imperator"

As noted elsewhere in this issue, it is but natural that much interest should be shown in such a large and elaborately equipped vessel as the "Imperator," and that curiosity should be excited by the vessel being ordered into dry dock for extensive repairs after seven months of service. As all who read the daily papers know, there have been many rumors as to the real reason why the vessel was taken out of service. While the American office of the ship's owners emphasizes its lack of knowledge concerning the vessel's troubles, the home office cables protesting denials of all rumors.

As Hamlet said of the lady, the company protests too much, we think. Surely there is nothing so very seriously wrong with this object of universal interest. The curiosity of engineers is not morbid; it is professional. Perhaps the German office does not appreciate the interest American engineers take in this ship. If it did, it would enlighten and not mystify.

One of the most beneficial developments that the power plant can sustain will be passing of the "fireman" and the coming of the combustion engineer, as has already been the partial replacement of the engine tender by the engineer.

That is a good suggestion, Mr. President, from the American Institute of Consulting Engineers, that one or more engineers be appointed upon the Interstate Commerce Commission. The time is ripe for the recognition of the worth of an engineering training for positions of this kind, and the profession contains men who would bring to the commission dignity and poise as well as real ability. It must be refreshing, too, to have a suggestion for an appointment from a body of men who have no candidate to urge.

When the recommendation is openly made in the forum of the American Society of Mechanical Engineers that the legislatures of the several states pass uniform laws governing the construction, installation and operation of steam boilers, and when the American Boiler Manufacturers' Association and the National Association of Tubular Boiler Manufacturers are actively advocating the passage of such laws it begins to look as if the legislators would have to sit up and take notice. What has become of the effort to get the National Commission for Uniform State Laws interested in the subject?

An appropriation which should find favor in the eyes of legislators who have the interest of the people at heart is that requested by the Department of Commerce (Bureau of Standards) for the investigation of the standards and methods of measurements, and for determination of the quality and cost of service required in the regulation by states and cities of public utilities, such as the electric light and power companies, city and interurban railway companies, telephone companies, gas companies, water companies and every public utility company, and for furnishing technical data and expert advice to state and city officials upon request in connection with the regulation or operation of public utilities, or of thefulfillment of municipal contracts for street lighting, water, gas or electric power, etc.

Readers with Something to Say

Discovered Large Crack in Boiler

Reading the article on boiler explosions and inspection in the Nov. 25 issue of POWER, page 746, reminds me of a near explosion in a plant where I was chief engineer.

On taking charge of the plant I found three boilers working under 90 lb. steam pressure, furnishing steam to a 14 and 26 by 42-in. tandem-compound engine operating a 2,000,000-gal. pump for water-works. On examining the boilers I found a slight leak near the lap seam, the steam issuing, apparently, from around the rivet. I asked the retiring engineer about it and he said the old boiler had been leaking for six months and he had nearly beat the head off the rivet trying to stop the leak. After a closer examination I found that the steam was coming through a crack in the boiler plate and at once had the boiler cut out of service, which brought the manager down on me in no uncertain terms, as the plant was seriously crippled without the boiler. I pointed out to him the danger of an explosion and he agreed to wire for an inspector, who arrived the next day. We put the boiler under a hydrostatic pressure of 90 lb. and a little water came out through the crack. The inspector struck the plate with his hammer and it opened for 5 in. right along the seam, and internal inspection was then made and seven crowfoot stays were found broken. A boiler maker was called in and after cutting out a strip of the plate a crack 28 in. long was found on the under side and hidden by the lap seam.

Just how long it would have been before the boiler would have exploded is a matter of speculation, but I think that any practical man will agree with me that it would not have been long.

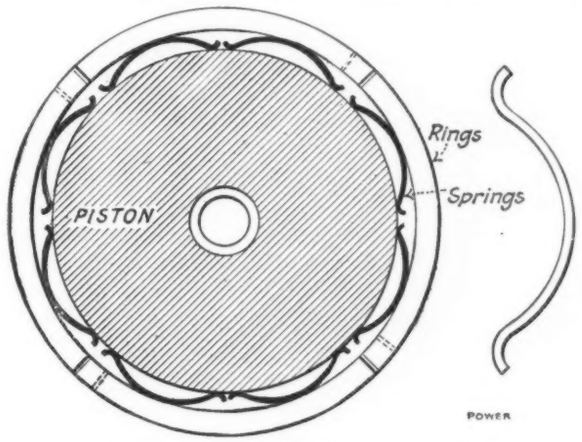
H. MCGIBONEY.

Middlesboro, Ky.

Piston Rings Wear Cylinder

Concerning the trouble A. M. Robb is having with pump-piston rings wearing, as told in the Nov. 11 issue, page 692, it is not stated whether the rings are cast iron or not, nor whether the lining of the cylinder is brass which would have a great deal to do with its wearing qualities. The pressure, 160 lb., is too high for packing, as it will cause undue friction and cutting, especially if the rings and lining are of the same material. This case is different from a steam cylinder in that there is more or less grit present in the water, and that there is no oil used to lubricate the rings as is done in a steam cylinder. I would suggest that the edges of the ring be well rounded off, and that the holes be all plugged up and flat springs be used under the rings as shown in the accompanying sketch. This ring is made in four segments with the ends lapped, and with two or more springs under each segment. The springs are long enough to entirely fill the space with a short dowel pin between each one to prevent them from getting out of place. This method of packing gives an even pressure on the entire surface, and can be made strong or weak as desired. The ring should be of a different metal than the cylinder lining, as it will give less friction than two metals of the same kind.

There are some steam engines the pistons of which have rings set out by the steam pressure, but the cylinder requires more oil and the wear on the ring and cylinder is uneven in the cylinder as the pressure on the ring is greater at the beginning of the stroke than it is after cut-



SECTION OF PISTON, SHOWING RINGS

off takes place. The pressure in the steam cylinder would be less against the ring than in the pump, and I do not believe the type of piston he mentions will give satisfactory results in a water cylinder.

J. C. HAWKINS.

Hyattsville, Md.

Slippage of Reciprocating Pumps

The article by M. K. Baer, on the above subject, appearing in the Apr. 8 issue, and the replies thereto, in the June 3 issue, have just come to my attention. The synopsis of this article says that it is folly to assume a general figure of 5 per cent. as the slippage of reciprocating pumps and the slip may be anything from 5 to 85 per cent. Thanks are due Mr. Baer for having brought up this question for discussion and it is to be hoped that other opinions will follow.

My opinion is that slip can only be stated for a pump in good condition. This is in line with the rule followed in making other tests. Nobody would think of basing the strength of some material on a defective test bar. Defects in design, material, workmanship, or due to wear are exceptions to the rule and must be remedied, but not pointed out as normal. Mr. McMillan expresses the same sentiment in saying that by keeping a pump in proper repair the slippage can be kept within a range of from 2 to 5 per cent. throughout the year.

The tests given in the article all show high slippages such as should not have been tolerated for any length of time. The 16 and 9 by 12-in. fire pump is stated as having a clearance of 0.02 in. Good workmanship demands

that the clearance of a plunger of this size should not exceed 0.001 in. in diameter or $^1/_{20}$ of the actual play. This would have reduced the slip to some reasonable amount.

No description is given of the 7½ and 4½ by 6-in. pump, so I assume that it was a pump with an improperly packed piston. To pack a piston just right is a difficult job. It must be tight enough to prevent leakage and yet not so tight as to produce undue friction. It is difficult to judge from outside appearances when this condition prevails. A packed piston may look nice on a drawing and be theoretically perfect, but in practice it is frequently far from this condition. A plunger and ring, on the other hand, is theoretically slightly imperfect and is looked upon by many with contempt for this reason. In practice, however, it is not any more imperfect than the packed piston and can be depended upon to last for a long time. So while a newly and properly packed piston pump may show less slip than a plunger and ring pump, the slip will rapidly increase while in the latter it will remain constant. The slip, therefore, largely depends upon the condition and cannot be stated by any rule or formula.

As mentioned in the article the slip is also dependent upon the speed, since the rate of leakage is practically constant and if expressed in per cent. of the capacity will be greater at reduced capacity. If the slip at full capacity is 5 per cent., then at half capacity it will be 10 per cent., and by continuing to reduce the speed a point will be reached where the slip is 100 per cent. This law is utilized in testing small pumps. The discharge valve is closed and sufficient steam pressure is admitted to obtain the normal water pressure. If the speed is then about 1 per cent. of the normal, the pump is passed as satisfactory. Thus a 6 and 4 by 6-in. pump, the normal speed of which is 65 ft. per min., will pass the test if it does not run much over 0.65 ft. per min. with closed discharge. For the sake of convenience this speed is expressed by the number of single strokes per minute and is called "lost actions." A pump with a 6-in. stroke makes 260 strokes per minute and will therefore pass with three lost actions. Here is a pump in perfect condition showing a slip varying from 1 per cent. at full load to 100 per cent. at $^1/_{100}$ load. Pumps handling large amounts of water have frequently shown no slip or even negative slip, which means a volumetric efficiency (1 — slip) of over 100 per cent. This was due to the fact that the water continued to flow after the plunger had stopped. The range of slippage could therefore be extended considerably and be made to look still more formidable.

Mr. Newbold's formula does not seem to agree with practice. The accompanying table calculated by this formula shows that pumps of 4-in. stroke and under do not pump any water at all, while as a matter of fact thousands of them are in successful operation. The formula is

$$x = \frac{\sqrt{P} \times 2000 \times G}{L \times S}$$

where

x = Percentage of slip;
P = Pounds pressure of the liquid;
G = Specific gravity of the liquid;
L = Length of stroke in inches;
S = Piston speed in feet per minute.

With 150 lb. as the pressure of the water and the specific gravity 1, the formula reduces to

$$x = \frac{24{,}500}{L \times S}$$

Using this formula the various percentages of slippage given in the table were obtained.

SLIP AS OBTAINED FROM FORMULA

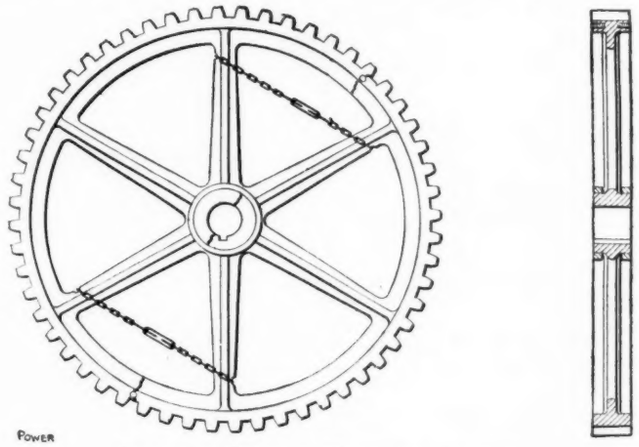
L Stroke, in.	S Speed, feet	x Per Cent.
3	40	204
4	50	122
5	60	82
6	65	63
8	75	41
10	80	30.6
12	90	22.7
15	100	16.3
18	110	12.4
24	120	8.5
36	135	5
48	145	3.5
60	150	2.7

F. F. NICKEL.

East Orange, N. J.

Quick Repair to Broken Wheel

Recently a serious breakdown occurred to one of our electric hoists that hauls coal up from one of our dip entries. While running at full speed around a curve the wagons left the track and ran into an obstruction, breaking the rope and at the same time breaking the large gear wheel on the drum shaft, completely in two, the pieces falling into the wheel pit. The accident happened while making the last trip, and the broken wheel was not discovered until the night shift went on duty. The hoist is nearly a mile and a half from the pitmouth, and as we had no duplicate of this wheel and could not afford to



CRACKED WHEEL HELD BY CHAINS

lose much time, we had the pieces brought out to the blacksmith shop, and laying them on the floor, we leveled them up so that the fractures would come well together. We then put a piece of ½-in. chain with a 1-in. turnbuckle around two of the arms on each side of the hub as shown, cutting a nick on the arms to prevent the chains from slipping. We then shrunk a strong ring on each side of the hub. This pulled the wheel together until the cracks were hardly perceptible. We drilled some holes in the rim of the wheel right in the cracks, tapping the holes for ¾-in. bolts, on both sides of the wheel and later pulling the bolts up tight to prevent any side movement of the parts of the wheel.

The dimensions of the wheel are as follows: Diameter, 68 in.; width of face, 6 in.; length of hub, 8 in.; bore, 5

in. It was nearly midnight when we got the wheel outside the mine and we had it repaired and back in place again in time for the next day's run, which we consider a quick repair job. The wheel is still running and has not given any signs of weakness, although it has been in operation for over six months.

T. D. PARFITT.

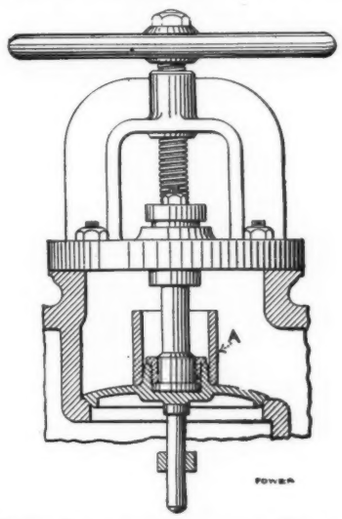
Claridge, Penn.

Reducing Valve-Stem Wear

The sketch shows how I stopped the chattering of the disk of our 8- and 6-in. globe valves which had worn off the ends of the valve spindles, so severe was the chattering. We have had five shutdowns in parts of the plant, owing to accidents of this nature.

Various means of stopping the chattering have been tried with varying degrees of success, but the means herewith described has proved the best.

The valve stem was screwed out of the hood and a piece of 2-in. pipe *A* about 1½ in. long placed on the stem between the hood and the top of the disk, the pipe going



PIPE COLLAR PREVENTS WEAR OF VALVE STEM

over the nut that holds the disk to the stem. When the valve is opened the stem is screwed out so that the collar of pipe on the stem is held tightly between the bonnet and the disk. This, of course, prevents any movement of the disk and consequently the end of the stem is not worn and there is no more chattering.

HARRY BIEHL.

Philadelphia, Penn.

Water in Engine Cylinders

The article describing the experiments of H. Bollincks with water in engine cylinders, as published in the Aug. 26, 1913, issue, reminded me of my own experience.

After warming a 14x30-in. Corliss engine, which I had just taken charge of, I started it slowly, but it had not made more than ten revolutions when there was a loud report in the cylinder followed by heavy blows at each stroke. Investigation revealed that water had accumulated in the exhaust pipe which was not provided with a drain as it formed a trap or pocket between the heater and engine cylinder, and caused the piston rod to punch through the follower plate.

I believe that if the engine had been started quickly it would have discharged the water from the cylinder without doing damage. The reason I believe this is because of another experience related below.

The fireman had flooded the boiler and much water was being carried to the engine. At the time I was on the second floor of the factory repairing a line shaft when my attention was called to the slugs of water coming from the end of the exhaust pipe. Rushing to the engine room, I shut down and was surprised to find that no damage had been done.

Paterson, N. J. PETER VANDENBERG.

Graphite Reduces Scale Formation

Having read the various articles and comments on the use of graphite in steam boilers, I am moved to relate my experience.

The scale evil has always been serious with us, and seemed nearly impossible of prevention. We have two 120-hp. horizontal return-tubular boilers in continuous service from June until Sept. 20, both boilers being run at full-rated capacity. The conditions make this service necessary. As an illustration of the amount of scale taken from the boilers after such runs, my records show that at the end of the 1912 season, 350 lb. of scale was removed from each boiler.

On June 16 and 27, 1913, Nos. 1 and 2 respectively, were put in service, each having graphite fed in but not strictly according to directions. No. 2 boiler gave 25½ lb. of scale when it was cleaned on Oct. 14, and No. 1 gave 49 lb. There still remains a little scale in each boiler which is expected to come off at the next cleaning.

Fearing that the scale might be thrown down in such quantities as to bag the shell, I fed in daily only one-fourth the amount specified in the directions.

FRANK C. B. SPEARE.

Cape May City, N. J.

Fixing the Blame

The editorial with the above title in POWER of Nov. 18 suggests a condition worthy of mention. Some four years ago considerable trouble was had by a company in operating its boilers. The tubes were continually being replaced, because of corrosion. At length the makers of a boiler compound overcame the trouble by supplying a barrel of their compound, and everything went on smoothly for about two years. Then the company hired another superintendent.

This man, a young college graduate, stopped the use of the compound and began using graphite. In a year's time, when the annual internal inspection became due, one of the boilers was so badly corroded at the tube headers and the nipples that the insurance company canceled its policy and condemned the boiler.

The owners are still operating this boiler, however, and while they are using a much more expensive compound, it is not doing any good. Furthermore, it is believed that the other boilers in the plant will be condemned at the next inspection. Here is a case where there should be some authority to which to appeal before the condemned boiler explodes and perhaps destroys many lives.

Hazleton, Penn. A. G. JAMES.

Inquiries of General Interest

Load Factor—What is meant by the term "load factor?"

J. W.

The generally accepted definition of "load factor" and the one adopted by most large power companies in this country, is the ratio of the average power to the maximum power during any given period.

Emery Around a Dynamo—Why should emery-cloth or paper not be used on a dynamo commutator?

W. N. J.

Emery is objectionable around a dynamo because its loosened grains are likely to get in the bearings and start cutting. It is better to use fine sandpaper as any loose flint getting into the bearings becomes ground up and discharged and is much less likely to set up cutting of the bearings.

Gas Heat Value—Is 120 B.t.u. per cu.ft. a good heat value for producer gas? In such a statement, the gas is presumed to be under what pressure, for the heat value per unit of volume must vary with the pressure?

H. H.

A value of 120 B.t.u. per cu.ft. of producer gas is low, as average producer gas would be about 140 B.t.u. per cu.ft. Unless otherwise qualified, statements of heat value per cubic foot are based on the density of the gas when at atmospheric pressure.

Boiler Compound—What is the composition of the Navy Standard Boiler Compound?

G. H. B.

This is a powdered compound composed of at least 64 per cent. of anhydrous sodium carbonate (Na_2CO_3), 15 per cent. of tri-sodium phosphate (Na_3PO_4 12 H_2O), 1 per cent. of dextrine or starch and 10 per cent. of a tannin compound such as mangrove bark, cutch or catechu, containing at least 35 per cent. of tannic acid. By thorough digestion these materials are intimately united, then dried, finely powdered and well mixed.

Injection Water for Oil Engines—What is the purpose of injecting water into the cylinder of an oil engine?

J. V. H.

For suppressing the force of the explosion and for permitting a higher compression without danger of preignition. In some instances, this water is injected into the cylinder separately; in others, where a light oil such as kerosene is used with a carburetor, water is sometimes injected along with the mixture from the carburetor. There is still another purpose of water injection, which was devised by Prof. Hopkinson, and described in "Power" for Spt. 2, 1913, wherein the water takes the place of the cooling jacket, and is impinged in slugs upon the head of the piston and on the exhaust valve.

Increasing Weight of Governor Balls—What effect has increasing the weight of the fly balls of a simple pendulum governor?

F. D.

Neglecting the weight of the arms, links and collar, which is small usually compared with the weight of the balls, increasing the weight of balls has no effect on their position for a given speed and consequently has no effect in changing the speed of an engine. The weight of the collar tends to keep the balls lower and retard the action of the governor. When the collar carries a weight, then an increase in weight of the governor balls increases the lifting power of the governor and has the same effect as taking load off the collar; i.e., the balls rise to cutting-off position of the governor for less speed and the engine is regulated to a slower speed.

Relative Coal Consumption—Why should more coal be used for driving our flour mill with an 18x42-in. Corliss engine during the night shift of 12 hours than when using the same engine and a 12x30-in. Corliss during the 12-hour day shift? The engines are belted to separate loads.

J. F. M.

The actual load and the uses of steam during the night shift may be greater than during the day shift. The facts should be ascertained by indicating the engines and inspecting all drafts of steam from the boilers and noting all operating conditions, each at intervals of about 20 min. during at least 24 hours. If such observations do not account for the discrepancies, they may be due to starting the day shift with high water and fresh fires and starting the night shift with low water and low fires and charging a disproportionate amount of coal to the night shift; or there may be a difference in firing.

Cost of Electric Hoisting—An electric crane lifts 3000 lb. 37 ft. Neglecting friction, motor efficiency and power consumed on the down trips, what would be the cost of handling 720 such loads, with current costing 3c. per kw.?

J. W. B.

One kilowatt-hour is equivalent to 2,654,200 ft.-lb. per hr. The net energy required to lift one load of 3000 lb. 37 ft. would be

$$3000 \times 37 = 111{,}000 \text{ ft.-lb.,}$$

and, therefore, a service of 1 kw.-hr. would raise

$$\frac{2{,}654{,}200}{111{,}000} = 23.9 \text{ loads}$$

At 3 cents per kw.-hr. the cost of current per load would be

$$\frac{\$0.03}{23.1} = \text{about } \$0.0013$$

and 720 such loads would cost

$$720 \times \$0.0013 = \$0.93\tfrac{6}{10}.$$

Gas-Meter Correction—Will a gas meter, calibrated for a pressure of 1½ in. of water, pass three times the quantity for every cubic foot registered when the actual pressure is 4½ in.?

H. H.

No. Gas pressure, stated in inches of water like gage steam pressure, is the pressure above the atmosphere. The latter being 14.7 lb. per sq.in. abs. (i.e., above vacuo), is equal to a pressure due to a column of water,

$$33.947 \times 12 = 407.364 \text{ in.;}$$

hence the absolute pressure equal to 1½ in. water pressure above atmosphere would be

$$407.364 + 1.5 = 408.864 \text{ in. of water abs.}$$

At 4½ in. water pressure above atmosphere, the absolute pressure would be

$$407.364 + 4.5 = 411.864 \text{ in. of water, abs.;}$$

so that one cubic foot at 4½ in. water pressure above atmospheric pressure would be equal to $\frac{411.864}{408.864}$ or 1.0073 as much as at 1½ in. water pressure above atmosphere, i.e., at 4½ in. water pressure there would be not quite ¾ of one per cent. more weight of gas per cubic foot than at 1½ in. water pressure.

Tube Baffling—Is it necessary to cover the top and lower side of the bottom row of tubes in a Heine boiler and the same of the top row of tubes, i.e., should these tubes have tile completely around them?

E. F. M.

Where such a boiler is fired by mechanical stokers or with a forced draft or both, for the better protection of the lower row of tubes, the baffle bricks are made to completely encircle these tubes. But where hand firing and natural draft are used, it has been found advisable to cover only the upper side of the lower row of tubes, especially where the coal tends to form stalactites, that is, deposits hanging from the brick, because in cleaning off the lower side of the tubes, or in the operation of hand firing, the undersides of the bricks, when exposed, are likely to be broken. This would allow the fire to go up in between the bricks like a "blow-pipe flame" causing excessive heat or local burning, which is detrimental to the tubes. The firebrick on the upper row of tubes is intended to act only as a baffle and at a point where the fire flame is not so intense, and where the brick is not likely to be disturbed so that only the upper side of the upper row of tubes needs to be covered. Hence tile forms completely surrounding the upper row of tubes are unnecessary and would detract from the heating surface of the boiler.

Engineers' Study Course

Trigonometry —VII

Last Lesson's Answers

26. If $sin\ 45° = \frac{1}{2}\sqrt{2}$ the sine of 22 deg. 30 min., or half of 45° is (from equation 23, last lesson)

$$sin\ 22° 30' = \sqrt{\tfrac{1}{2}(1 - cos\ 45°)}$$

As the sine and cosine of 45° are the same

$$sin\ 22° 30' = \sqrt{\tfrac{1}{2}(1 - \tfrac{1}{2}\sqrt{2})} = 0.3828$$

From equation 24

$$cos\ 22° 30' = \sqrt{\tfrac{1}{2}(1 + cos\ 45°)} = \sqrt{\tfrac{1}{2}(1 + \tfrac{1}{2}\sqrt{2})} = 0.9239$$

From equation 25

$$tan\ 22° 30' = \sqrt{\frac{1 - cos\ 45°}{1 + cos\ 45°}} = \sqrt{\frac{1 - \frac{1}{2}\sqrt{2}}{1 + \frac{1}{2}\sqrt{2}}} = 0.4142$$

From equation 26

$$cot\ 22° 30' = \sqrt{\frac{1 + cos\ 45°}{1 - cos\ 45°}} = \sqrt{\frac{1 + \frac{1}{2}\sqrt{2}}{1 - \frac{1}{2}\sqrt{2}}} = 2.4137$$

(Note that these values lie about midway between those given for the corresponding functions of 22° and 23° in the table in Lesson II.)

27. Required to find an expression for $sin\ (A + B + C)$. Calling $A + B$ for the time being one angle, then, from equation 11,

$$sin\ [(A + B) + C] = sin\ (A + B)\ cos\ C + cos\ (A + B)\ sin\ C$$

Also from equation 11,

$$sin\ (A + B) = sin\ A\ cos\ B + cos\ A\ sin\ B$$

and from equation 13

$$cos\ (A + B) = cos\ A\ cos\ B - sin\ A\ sin\ B$$

Then, substituting in the first equation,

$$sin\ [(A + B) + C] = (sin\ A\ cos\ B + cos\ A\ sin\ B)\ cos\ C + (cos\ A\ cos\ B - sin\ A\ sin\ B)\ sin\ C$$

or

$$sin\ (A + B + C) = sin\ A\ cos\ B\ cos\ C + cos\ A\ sin\ B\ cos\ C + cos\ A\ cos\ B\ sin\ C - sin\ A\ sin\ B\ sin\ C$$

28. From equation 5 (Lesson III)

$$sec\ (A + B) = \frac{1}{cos\ (A + B)}$$

and, from equation 13

$$cos\ (A + B) = cos\ A\ cos\ B - sin\ A\ sin\ B$$

therefore

$$sec\ (A + B) = \frac{1}{cos\ A\ cos\ B - sin\ A\ sin\ B}$$

This may be reduced to terms of the secant and cosecant if desired as

$$sec\ (A + B) = \frac{sec\ A\ sec\ B\ csc\ A\ csc\ B}{csc\ A\ csc\ B - sec\ A\ sec\ B}$$

29. *Sin* 3 *A* is the same as $sin\ (2A + A)$, then, from equation 11,

$$sin\ (2A + A) = sin\ 2A\ cos\ A + cos\ 2A\ sin\ A$$

Substituting the values of $sin\ 2A$ and $cos\ 2A$ (equations 19 and 20)

$$sin\ (2A + A) = 2\ sin\ A\ cos\ A\ cos\ A + (1 - 2\ sin^2 A)$$
$$sin\ A = 2\ sin\ A\ cos^2 A + sin\ A - 2\ sin^3 A$$

From equation 1 (Lesson III).

$$cos^2 A = 1 - sin^2 A$$

then

$$sin\ (2A + A) = 2\ sin\ A\ (1 - sin^2 A) + sin\ A - 2\ sin^3 A$$

which simplifies to the form

$$sin\ 3A = 3\ sin\ A - 4\ sin^3 A$$

30. Given $tan\ 30° = \frac{1}{\sqrt{3}}$.

From equation 21,

$$tan\ 60° = \frac{2\ tan\ 30°}{1 - tan^2\ 30°} = \frac{2 \times \frac{1}{\sqrt{3}}}{1 - \left(\frac{1}{\sqrt{3}}\right)^2} = \frac{\frac{2}{\sqrt{3}}}{1 - \frac{1}{3}}$$

$$= \frac{3}{\sqrt{3}} = \sqrt{3}$$

As $tan\ 30° = cot\ 60°$ (60° *and* 30° being complementary angles) $cot\ 60° = \frac{1}{\sqrt{3}}$.

From equation 7 (Lesson III)

$$sec^2\ 60° = 1 + 3 = 4$$
$$sec\ 60° = \sqrt{4} = 2$$

From equation 5

$$2 = \frac{1}{cos\ 60°} \qquad cos\ 60° = \tfrac{1}{2}$$

From equation 2

$$\sqrt{3} = \frac{sin\ 60°}{\frac{1}{2}} \qquad sin\ 60° = \tfrac{1}{2}\sqrt{3}$$

Solution of Right Triangles

Recalling the definitions of the functions of an angle, as given in Lesson I, it begins to be evident how it may be true, as stated, also in the first lesson, that if any three of the six parts (three angles and three sides) of a triangle are known and at least one of them is a side, the shape and size of the triangle may be determined, at least insofar as this applies to right-angled triangles.

To recapitulate the relations between the sides and the trigonometrical functions of the oblique angles of a right triangle:

$$sine = \frac{opposite\ side}{hypothenuse} \qquad cosecant = \frac{hypothenuse}{opposite\ side}$$

$$cosine = \frac{adjacent\ side}{hypothenuse} \qquad secant = \frac{hypothenuse}{adjacent\ side}$$

$$tangent = \frac{opposite\ side}{adjacent\ side} \qquad cotangent = \frac{adjacent\ side}{opposite\ side}$$

Suppose it is given that the base and altitude of a right triangle are b and a, respectively. Here three parts are given, the right angle and two sides. The other side (the hypothenuse) can be found as the square root of the sum of the squares of the other two sides or

$$c = \sqrt{a^2 + b^2}$$

The sine of one oblique angle then will be $\frac{a}{c}$ and the sine of the other $\frac{b}{c}$. From the table (Lesson II) both of these angles may be found, or, one being found from the table, the second can be found by subtracting the first from 90° (the oblique angles of a right triangle being complements of each other, as previously explained).

Suppose the hypothenuse of a right triangle and one side are given as c and b respectively. Again, three parts are known. The third side will be

$$a = \sqrt{c^2 - b^2}$$

The sine of the angle opposite to the side b will be $\frac{b}{c}$ from which the angle B may be determined and the third angle (A) will be

$$90 - B = A$$

Suppose one oblique angle of a right triangle is given, A, and the adjacent side is b. Three parts are given, the third being the right angle. From the table *cos A* may be found, and the cosine being equal to the adjacent side divided by the hypothenuse, the latter can be found, as

$$c = \frac{b}{\cos A}$$

Two sides are now known, b and c; the third side a and also the angles may be found as before.

Study Questions

31. In a right triangle the sine of the acute angle at the base is ½ and the hypothenuse is 36 in. What are the lengths of the other sides and what is the sine of the other oblique angle?

32. The base and altitude of a right triangle are 20 and 32. Find the angles and the hypothenuse.

33. In a right triangle the hypothenuse is 235 and one angle is 43 deg. Find the other acute angle and the two other sides.

34. The angle of elevation to the top of a chimney is 47 deg. as measured from a point 180 ft. horizontally from its base. What is the height of the chimney?

35. When the 8-in. crank of a horizontal engine is vertical the piston is 1½ in. past midstroke. At this instant, what angle is the connecting-rod making with the guides?

Edinburgh to Have New Central Station—The electric-lighting committee of the Edinburgh Municipal Corporation, which has the monopoly of the production and distribution of electric current in Edinburgh, is preparing plans for a new generating station. The existing plant, which has been operated so successfully for many years that the committee has gradually reduced prices to 2¼c. per unit for manufacturing purposes (1⅜c. to large consumers) and 5¼c. per unit for domestic lighting (with discounts which bring it down to 4c.), will not economically produce enough current for the requirements of the near future. In arranging for another station, all new ideas, designs, and appliances will be considered by the committee, with the purpose of supplying electricity at the lowest possible figures.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

What is "the largest" anyhow? We said the new Equitable Building was to have the largest number of elevators (48) because we so read it in "Building Management." Now Charley Naylor pooh-poohs us and intimates the Equitable is a bungalow alongside the Marshall Field Co. store, which has 64 passenger elevators, 14 freight, and 21 dummies and package hoists. And by way of good measure, he says the annex to Field's has 15 passenger, two freight and six dummies. Ring the bell if you want to go up!

However, some large things were done last year, as is pictured in this week's foreword. That young "1914" recorder is no great shakes for infantile beauty, but get that enthusiasm!

In a barefaced attempt to exploit our erudition last week, we said we knew two Latin quotations, and quoted one of them, "Caveat emptor" (Let the buyer beware). Then "Poor Old Bill" leaves his domiciliary elevation to dare us to tell the other one. No Trentonian can give us the dare, and get away with it—the other one is "Gesundheit"!

A Brooklynite has reached the tender age of 102 by indulging in "three good laughs an hour." We know where we could procure two good smiles an hour if we needed them, but three good laughs are hard to find. Possibly some reader may know where Brooklyn is, and help us out.

"Conservation of natural resources is," remarks the Buffalo "Express," "talking forever about water power and coal mines, and never using them." First thing you know, Mister "Express," somebody will make you eat them crool words.

"A mechanic was ordered to take care of a small gasoline pumping outfit during the night as we were short of water," relates an American engineer in Cuba. "About 12 o'clock the chief mechanic, while making his rounds, missed the sound of the exhaust. Upon investigation, he found that the connecting rod had worked loose and that piston and rod were reposing across the room and the engine a beautiful wreck. The boy was still asleep at his post."

Figures 2 ft. high are to take the place of street names on the Pittsburgh street cars. Don't know how this will work out, but what do New Yorkers know about street railways, anyhow? Two bells start the horses, and one stops them.

Owing to a severe shortage of water supply, a Kentucky electric-light company had to get its water by rail. You'd think that in the Blue Grass country water was running about loose, but this was boiler-feed water, not drinking.

A HAPPY NEW YEAR TO YOU

By B. Spills

We've said goodby to 1913 and howdy to 1914. Now let's be kind to the youngster; he's sturdy enough if we take the artist's conception of him in this week's foreword.

Big business enterprises are on the slate, the Federal and state governments are going to do things in our field, and others, and it's up to you and to me to sit tight, and look pleasant and hopeful.

If you read "Power's" editorial, "Progess in 1913," page 19, you must be startled by how much can be accomplished in one short year. My goodness gollies, old Pop Time sure can swing that grass-hook! Not so fast, Pop; we're missing some of the stations, and we'd like to stop off before our mileage runs out. (Pop's the only man who ever got away with the perpetual-motion idee.)

It's good to know, too, that the hearts of the big-wads have opened up the past year. For one thing, $550,000,000 are on tap for education and charity. We need all the education we can get, no matter who helps us to it. As for charity, we need that, too—in our hearts—for our fellows, our neighbors, and ourselves.

Here's for a happy new year!

Two Years' Experience with CO_2 Recorders

By Ernest O. Hickstein

SYNOPSIS—By stopping up all leaks in the settings and around doors, and by stimulating competition among the firemen, the average percentage of CO_2 was raised in nine months from 8 to 12.5 per cent.

It is rarely that one sees reports showing in detail just how boiler efficiencies have been increased, or by just what means low CO_2 percentages have been raised through the use of recorders. The writer has been in daily touch with one make of CO_2 recorder for two years past, the larger part of this time in a boiler plant where 20 recorders are in constant operation. The boiler house furnishes steam for power and industrial purposes for a large manufacturing plant on the New Jersey side of the Hudson River, and contains 20 water-tube boilers—14 Babcock & Wilcox and six Edgar—all connected with Green economizers. There are two large stacks, 250 ft. high. The boilers are hand fired, and dutch-oven furnaces extend 5 ft. in front of the boilers. A mixture of 90 per cent. No. 3 buckwheat and 10 per cent. soft coal is burned on a special small-hole dumping grate with forced draft. The maximum pressure in the ashpit is about 1¼ in. All the drums connect into one header. The boilers run day and night, usually at from 30 to 40 per cent. overload, and the demand for steam is regular. A CO_2 recorder unit and a recording thermometer are attached to each boiler.

The first month after the CO_2 recorders were installed, the average percentage of CO_2 was found to be around eight. By means of the methods to be recounted, this average was raised in eight months to 12.5 per cent. There has been little difficulty since, in keeping the average around that figure, but this seems to be about the limit under the conditions. The following tabulation gives the average for the plant during the eight months it took to raise the percentage up to a satisfactory figure and shows how gradual the improvement was:

	Per Cent.
1st month	7.78
2nd month	8.98
3rd month	9.26
4th month	9.88
5th month	10.24
6th month	10.59
7th month	10.73
8th month	11.65
9th month	12.46

How this was done can be told most directly by putting each step under one of three headings, which will be taken up in the following order:

1. Making the boiler settings tight.
2. Making sure that the recorder was getting a fair sample.
3. Improving on the methods of firing.

Making the Boiler Settings Tight

The boiler settings, when the recorders were put in, were in only fair condition. While some effort was made to prevent too much air leaking in, still if a match or candle were run along the brickwork, here and there, the flame would be sucked in. The black spots on the whitewashed brick, caused by little explosions in the furnace forcing soot out through the leaky settings, attracted attention primarily because it meant another coat of whitewash for the brickwork.

There had been no incentive to having the settings absolutely tight. No one knew just how much air leaked in and the Orsat remained in its corner for a year without being touched. But with the machines installed, things were different. The superintendent, on his trips through the boiler house, could not help seeing the CO_2 charts, which told him that things were not as well as they might be. Moreover, the recorders indicated whether conditions were improving; they supplied the incentive to the men in charge.

The brickwork of the settings, economizers, breechings and flues was overhauled, loose brick, wherever found, were replaced, and the boiler walls were well covered with asbestos cement. Care was taken that there was no leakage at the tube-blowing doors, or at the side, rear or firing doors. The tops of the boilers were covered, 1½ in. deep with cement over the brickwork. Of course, these are ordinary precautions, but two things done, however, were more uncommon, and deserve special mention.

As the boilers were often worked steadily for several months at a time, and at overload, cleaning the cinder pits by hand was arduous. Shortly before the recorders were installed, a hole about a foot square, had been drilled at the center of each of these pits, through to the cellar. The idea was that cinders carried into the second pass of the B. & W. boilers and falling into the cinder pits could be washed down into the cellar by playing a hose through the side doors. This idea worked very satisfactorily, except that such poor means were first used for keeping the holes air-tight, that considerable leakage resulted. When it became a question of getting high CO_2, however, a casting was inserted in each hole and cemented in place. These castings had grooves along the bottom to hold tight-fitting slides in place, and by this means all leakage through the holes was eliminated.

Leakage around the large front doors of a B. & W. boiler has a doubly bad effect, for in addition to diluting the hot flue gases with cold air, it strikes directly against the front header. These large heavy doors are hard to keep tight, so a simple but ingenious plan was adopted for preventing leakage. The air that leaks in through the front door passes either through the spaces between the headers, or up through the large opening just in front of the top of the headers; it was decided to block both these paths. Asbestos rope was forced into the openings between the headers with a calking tool. Also, a sheet-iron apron about 18 in. wide, and extending the width of the boilers was suspended just above the front doors, to seal the space above the header. Thereafter it was necessary only to keep the outer doors reasonably tight to prevent the air from cooling the front headers—the leakage into the boiler having been reduced to a negligible quantity.

Making Sure That the Recorder Was Getting a Fair Sample

Samples of the flue gas are taken by means of a ¾-in. pipe, inserted through the brickwork, directly in front of the breeching. This pipe is open at the end only. The recorders were installed under the supervision of a representative of the manufacturer, and at that time the ¾-in. pipes were only 3 ft. 6 in. long, probably because the alley between the boilers was only a little over 4 ft. wide. This permitted the pipes to extend only a little over 2 ft. inside the wall. The boilers are 16 ft. wide and the breeching 4 ft.; hence the sample was taken from a sort of dead pocket far to one side of the current of gas through the boiler into the breeching.

It was then ascertained that the recorder readings did not agree with the Orsat readings taken through a longer sampling tube extending halfway into the boiler. Also, the readings from the recorder with the short tube did not vary nearly so much with varying conditions of the fire as did the Orsat readings. Thereupon it was decided to use a sampling tube extending 8 ft. into the boiler. This was done and an immediate improvement in the charts resulted.

Improving the Methods of Firing

The boilers having been put in first-class shape, it was "up to" the firemen to produce good results. The firemen at the plant are practically all illiterate foreigners having no interest in their work beyond "putting in" the 12 hr. of their shift. Being without education, there was no hope of advancement, and with the shop unionized after a very recent strike, the firemen getting $2.75 for 12 hr. work and threatening to go out again for more money, it seemed like a hard problem to induce the firemen to get a good CO_2 record. There was no way of finding out readily how much coal a man shoveled, or how much water each boiler evaporated. This, of course, made it difficult to detect shirking, particularly at night, until the recorders were put in. Thereafter the charts showed who worked and who did not, but unwillingness on the part of the boss fireman to reprimand shirking firemen, and his disinclination to lay off a man except for a "better reason than just a poor CO_2 record," made matters just as bad as before.

There was no money to establish a sort of bonus system for the best CO_2 average, and there was no way of making the men do their best for fear of losing their jobs, so, as a last resort, and with but little hope of success, the attempt was made to create a feeling of rivalry between the firemen.

Each CO_2 chart was averaged separately for the day and night shift, and the result recorded under the fireman's name. The average for each fireman was taken at the end of each two weeks, when the shifts changed. A page of the firemen's record book is shown herewith. With the help of the information given underneath, this record is self-explanatory.

SAMPLE PAGE FROM FIREMEN'S RECORD BOOK

CO_2 Record

Two Weeks beginning Monday, May 12, 1913

Night Shift	Mon. 12	Tues. 13	Wed. 14	Thu. 15	Fri. 16	Sat. 17	Sun. 18	Mon. 19	Tue. 20	Wed. 21	Thu. 22	Fri. 23	Sat. 2A	Sun. 25	Average
560 Scullion	3 13.5	4 14	5 13.5	6 13	8 12.5	9 10 12.5	11 10 A	12 13 13	14 13 12.5	1e 12	3e 13	5e 12.5	6e A		12.91
546 Cannon	4 14	5 14	6 13	7 12.5	9 12	11 10 12.5	12 13 ×	14 13 12	1e 11.5	3e 12.5	5e 11	6e 13	2 13		12.58
564 Shultz	5 14	6 12.5	7 12.5	8 10 ×	11 10 11.5	12 13 ×	14 13 12	1e 12.5	3e 12	5e 12	6e 14	1 13	3 12.5		12.59
550 Barnetzkay	6 12.5	7 12	8 10 ×	11 10 11.5	12 13 12	14 13 11.5	1e A	3e ×	5e 11	6e A	1 A	2 12.5	4 ×		11.86
575 Jordan	7 13	8 10 10	11 10 11.5	12 13 12.5	14 13 12	1e ×	3e 11	5e 12.5	6e ×	1 ×	2 12.5	3 13	5 13		12.10
562 Zeman	10 12	10 12.5	13 12.5	13 13	1e 11.5	3e ×	5e 12.5	6e 13.5	1 12	2 12	3 12	4 13	6 12.5	12-hr. shutdown	12.42
554 Tiese	11 10	12 13	14 13 12.5	1e 12	3e 12	5e 13.5	6e 13	1 13	2 13	3 12.5	4 13	5 13	7 13		12.88
553 Oberhaus	12 13	14 13	1e 13.5	3e 13.5	5e 13	6e 14	1 ×	2 13	3 13	4 13.5	5 13.5	6 13	8 12		13.18
548 Publiskie	14 13	× 1e	× 3e	13 5e	12.5 6e	12.5 1	× 2	× 3	13.5 4	13 5	13.5 6	12 7	9 10 9		12.39
556 Muchinsky	1e 12	3e 12	5e 13.5	6e 11	1 12	2 12.5	3 13	4 13	5 13	6 11.5	7 12	8 12	11 10 12.5		12.31
572 Bakoonas	3e 11.5	5e 12.5	6e 13	1 12.5	2 12	3 12	4 12	5 13	6 11.5	7 12	8 9 11.5	10 9 12	12 13 ×		12.13
545 Contract	5e 13.5	6e 13	1 13	2 12.5	3 13	4 12.5	5 13.5	7 13	7 13	8 9 11.5	10 9 11	12 13 11.5	14 13 12		12.54
547 Staffon	6e 14	1 12.5	2 13	3 12.5	4 13	5 13.5	7 13	8 12.5	8 9 11	10 9 12	12 13 11	14 13 12	1e ×		12.50
563 Kutakoski	1 11	2 13	3 13	4 12	5 14.5	7 12	8 11	9 10 ×	10 A	12 13 10.5	14 13 11.5	1e 11.5	3e ×		12.00
565 Bellows	2 13.5	3 12	4 12.5	5 13.5	7 13	8 12.5	10 12	11 10 12.5	12 13 12	14 13 12.5	1e 12.5	3e A	5e 13		12.03

Explanation: The first column under each date shows the boiler the man is firing; the second column, the CO_2 percentage for the fireman. "A" means that the regular fireman was not working, and a cross indicates that the recorder was not working well enough to get the correct average.

For example, Jordan, check number 575, on Monday night, May 12th, fired boiler No. 7 and averaged 13 per cent. On Tuesday night he had boiler No. 8 and half of No. 10, and obtained 10 per cent. CO_2 on boiler No. 10. The CO_2 average of No. 10, which is divided between two firemen, is ignored in keeping the firemen's record.

SAMPLE LIST, SHOWING FIREMEN'S CO_2 AVERAGES

Night Shift:

Two Weeks ending Sunday, May 25th, 1913

Good:		Fair:	
553 Oberhaus	13.18	562 Zeman	12.42
560 Scullion	12.91	548 Publiskie	12.39
554 Tiese	12.88	556 Muchinsky	12.31
565 Bellows	12.63	572 Bakoonas	12.13
564 Schultz	12.59	575 Jordan	12.10
546 Cannon	12.58	Poor:	
545 Contract	12.54	563 Kutakoski	12.00
547 Staffon	12.50	550 Barnetzkay	11.86

A list was then made and posted in the boiler room, showing how the firemen stood for the two weeks. The men were grouped under the headings "good," "fair" and "poor." A sample list is herewith shown, corresponding with the page of the record book.

From the very outset, the posting of this list created more interest than was hoped for. The men under the heading "poor" have to stand for a good deal of chaffing, and the leaders are usually pleased over their position. An idea of how much interest is taken in the list can be gained from the fact that frequently men on the night-shift report for work half an hour early, on the afternoon when the list is posted. Once the drawer containing the firemen's record book was broken open, probably by some

fireman who could not restrain his curiosity any longer. Before the installation of the recorders, most of the firemen would shut off their blast the minute they thought they were not being watched, regardless of what the steam pressure might be at the time. After the lists were first put up, quite the reverse happened. It was necessary, then, to watch the firemen only to make sure that they did not keep on the blast, when the safety valve was blowing.

A most potent factor in keeping up the fireman's interest in his position on the list is the superintendent. He often looks over it, and has pointed out to him, the worst and best firemen. When the lists were first posted, several of the lazier firemen complained that the last man got $2.75 for his day's work just as the first man. About this time, it was decided to run on one less boiler, and as talk of another strike was dying out—winter was drawing near—these men were laid off for a week or two. That one lesson was sufficient for the whole fire room. The only reward given now to the leaders on the list is preference for Sunday work, when there is a partial shutdown on that day.

They have also learned that a "hole" in the fire means low CO_2 and they are always on the lookout. They are careful to cover their fires evenly, at the back and sides of the grate as well as at the front, so that large quantities of air cannot be forced by the blast up along the bridge-wall or side walls or through the spaces between the firing doors in the front. They know by experience just what blast in the ashpit is needed for the best results, and by means of a glass U-tube put on the boiler front to show pressure in the ashpit, they can hold the blast at the desired mark.

The average of the CO_2 charts is taken by inspection, this method being by far the quickest and giving very satisfactory results. Some of the firemen, in their desire to get a good chart, will not shut off their blast, even when the safety valve is popping. Because of this fact, it is necessary to consult the steam-pressure chart before averaging the CO_2. When the steam is blowing off, the firemen should not be given credit for very high CO_2, nor should the conscientious fireman, who shuts off the blast when the steam is high, be penalized for a low percentage at this time.

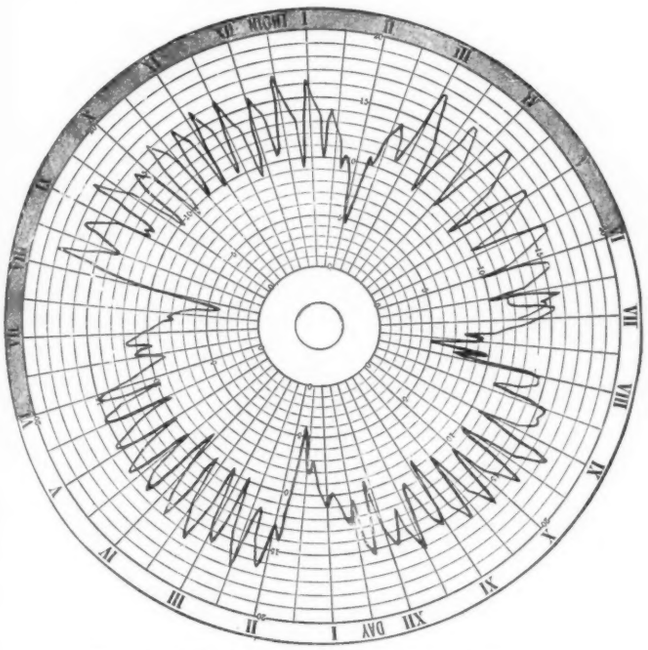


FIG. 1. CO_2 AVERAGE OF 14 PER CENT.

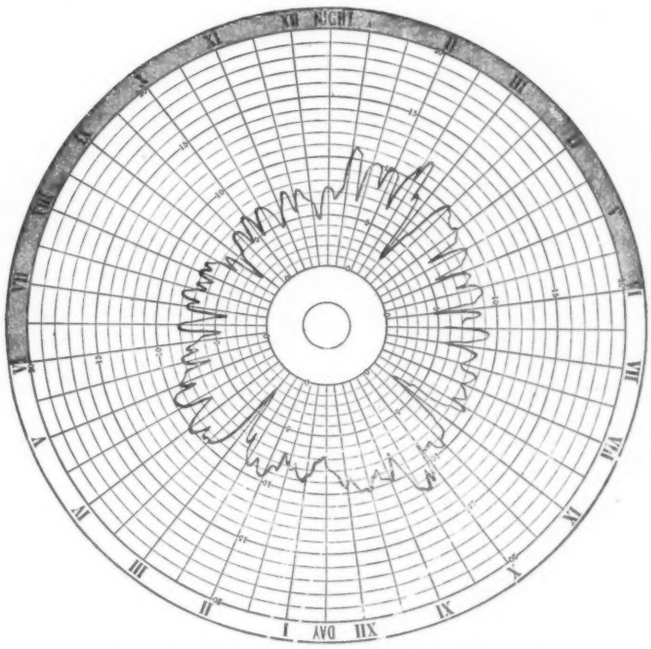


FIG. 2. CO_2 AVERAGE OF 8 PER CENT.

There are plants using CO_2 recorders, in which it is deemed best to conceal the record from the fireman who is making it. There is unquestionably a valid reason for this where a single recorder has to serve a number of boilers. By hiding the chart, the firemen cannot ascertain which of their number is being watched. Where there is a recorder for each boiler, however, much better results should be obtained by allowing the fireman to see the record he is making. If a fireman is anxious to make as good a chart as possible, he will watch the recorder as he covers his fire, cleans, etc. It will gradually be impressed on his mind that one method of doing this thing or that means a higher percentage of CO_2 than a second method, and slowly but surely, his whole system of firing will improve.

The firemen in the plant mentioned have found out for themselves that light firing at frequent intervals gives a higher CO_2 average than heavier firing at longer intervals.

CONCLUSION

It is the general impression that CO_2 recorders get out of order very easily and require considerable attention, an impression that is borne out by the experience of many engineers. The CO_2 recorder in a small plant is usually put in charge of the chief who is often the busiest man around the plant. He has his routine duties, subject to change in an emergency, but if the recorder is not working right, he will not call that an emergency, and may let it go for a day or a week. Consider how different it would be if the chief assigned one of his men to the instrument. A technical man is not needed to take care of it, although a certain amount of mechanical knowledge or instinct is very necessary. This man would be responsible to the chief for keeping the machine running, and would give attention to it at the proper time. This is unquestionably the only way in which a small plant can get its money's worth out of a recorder.

In a large plant, with a number of machines, there is usually one man specially engaged for the recorders. Whether the instruments pay will depend on this man; they make good if he does; they cannot, unless he does.

The occasional use of the Orsat apparatus to supplement the CO_2 record by a determination of oxygen and carbon monoxide is well nigh indispensable. Occasionally, when a recorder is showing an especially high CO_2 percentage, it is well to take an Orsat reading. The CO reading at this time might show the presence of sufficient CO to indicate excessive loss through incomplete combustion. Three-tenths of 1 per cent. of CO is a conservative maximum limit.

The efficiency for the first six months of the present year was slightly over 70 per cent.—4 per cent. higher than for the six months prior to the recorders' installation. A gain of 4 per cent., when the efficiency is 66 per cent. corresponds to a saving of 6.06 per cent. in the coal bill. This saving, based on an annual coal bill of $250,000, represents over $15,000 a year, or about $41.50 saved every day in the year, an amount which will pay the wages of the firemen.

Physical Properties of Fuel Oil

By J. J. McIntosh

SYNOPSIS—Weight, gravity and heat values and its use as a fuel compared with coal.

There are two kinds of petroleum fuel; that which has been worked over in a refinery, and that which is used in its crude or natural state. In the first state it is in the form of residue, the light distillates having been taken off. In this state but little variation will be observed in the properties of the oil, and it will be found to contain a small percentage of water. In its crude state, however, the oil is apt to show wide variations of gravity and impurities. If it is asphaltic, it will also contain notable amounts of sulphur, while the percentage of water is generally between 2 and 5 per cent.; though in some cases the water content may be as high as 10 per cent.

TABLE 1. PROPERTIES OF DIFFERENT OILS

Baumé Gravity	Specific Gravity	Weight Lb. per Gal.	Weight Lb. per Bbl.	B.t.u. per Lb.	B.t.u. per Bbl.
10	1.0000	8.33	349.86	18,280	6,395,441
11	0.9929	8.27	347.34	18,340	6,370,216
12	0.9859	8.21	344.82	18,400	6,344,688
13	0.9790	8.16	342.72	18,460	6,326,611
14	0.9722	8.10	340.20	18,520	6,300,504
15	0.9655	8.04	337.68	18,580	6,274,094
16	0.9589	7.99	335.58	18,640	6,255,111
17	0.9523	7.93	333.06	18,700	6,227,622
18	0.9459	7.88	330.96	18,760	6,208,810
19	0.9395	7.83	328.86	18,820	6,189,145
20	0.9333	7.78	326.76	18,880	6,169,229
21	0.9271	7.72	324.24	18,940	6,141,106
22	0.9210	7.67	322.14	19,000	6,120,660
23	0.9150	7.62	320.04	19,060	6,199,962
24	0.9090	7.57	317.94	19,120	6,079,013
25	0.9032	7.53	316.26	19,180	6,065,867

The principal constituents of crude petroleum are hydrogen and carbon, combined with small amounts of sulphur, oxygen and nitrogen. Sulphur has a low heat value and is otherwise an undesirable property of the fuel, while nitrogen may be classed as an inert impurity. Carbon contains 14,500, hydrogen 62,000 and sulphur 3900 B.t.u. per lb. If fuel oils were made up wholly of hydrogen and carbon, there would be almost an exact relation between the gravity and the number of heat units per pound, but the presence of varying amounts of other substances prevents this constant relation. It is found that oils rich in hydrogen are of lighter gravity and contain more B.t.u. per pound than oils that are rich in carbon.

In Table 1 will be found the weights and heat values of different oils. These data have been gathered from various sources, and have proved to be correct for the majority of cases, but as was noted above, the heat values vary to a certain extent in oils of the same gravity.

The oils found in the mid-continent and Eastern fields contain a paraffin base, while those produced in the Gulf and Western fields contain an asphaltum base. Generally speaking, the asphaltic oils are distinguished from the paraffin base oils by their greater specific gravity and viscosity. The asphaltic oils also contain large proportions of tar, pitch and bitumen, and therefore have relatively more carbon than the oils with the paraffin bases.

In studying the table it may be noticed that the average heat value of the oil, per pound, increases as the gravity diminishes. This would indicate that economy

TABLE 2. RESULTS OF TESTS ON BOILER BURNING FUEL OIL

Date of test	Mar. 11	Mar. 12	Mar. 13	Mar. 15	Mar. 15	Mar. 17	Mar. 18
Duration of test, hr.	7	7	7	5	2	7	5
Steam pressure, gage, lb.	185.5	185.5	187.1	186.6	185.5	186.2	186.5
Steam temperature, deg. F.	450.0	446.0	457.0	459.8	482.3	445.0	443.0
Superheat, deg. F.	68.0	64.0	72.0	84.0	107.1	63.0	61.1
Feed-water temperature, deg. F.	85.5	86.2	86.6	83.0	82.3	83.0	80.3
Factor of evaporation	1.22	1.2174	1.222	1.230	1.243	1.220	1.222
Flue-gas temperature, deg. F.	449.5	436.0	454.0	467.2	495.6	436.8	400.0
Draft in furnace, in.	0	0	0	0.010	0.050	0.020	0.013
Draft, top last pass., in.	0.034	0.010	0.074	0.128	0.250	0.010	0.030
Loss of draft through setting, in.	0.034	0.010	0.074	0.118	0.200	.....	0.017
CO_2 in flue gas, per cent.	14.53	14.38	14.4	13.57	13.14	14.54	14.21
Total water evaporation f. & a. 212 deg., lb.	113,581	94,781	118,64	111,744	48,230	96,476	46,350
Total water evaporation, actual, lb.	93,099	77,855	97,095	90,849	38,802	79,079	37,930
Total oil fired, lb.	7021	5738	7506	7411	3179	6146	2826
B.t.u. per lb. of oil, as fired	18,986	18,986	18,986	18,986	18,986	18,986	18,986
Fuel-oil temperature, deg. F.	131	139	131	118	118	136	139
Boiler hp. developed	470	392	491	653	699	399.5	268.7
Boiler hp. per cent. of rating	118.5	99.0	124.0	165.0	176.0	101.0	68.0
Actual evaporation per lb. of oil, lb.	13.26	13.57	12.93	12.25	12.20	12.86	13.42
Equivalent evaporation f. & a. 212 deg. lb.	16.18	16.52	15.81	15.07	15.16	15.68	16.40
Effi. boiler and furnace, per cent.	82.5	84.2	80.6	76.8	77.1	79.7	83.5

would be best subserved by the purchase of light oils; but since oil is sold by volume, instead of by weight, a barrel of heavy oil contains more heat units than a barrel of light oil. Thus, for example, a barrel of oil whose gravity is 10 on the Baumé scale contains 226,212 more heat units than one whose gravity is 20 on the same scale. The following formulas will be found convenient for converting gravity values:

$$Specific\ gravity = \frac{140}{130 + B}$$

$$Baume\ gravity = \frac{140 - 130}{sp.gr.}$$

It is found in practice that 1 lb. of oil is equivalent to 1.6 lb. of the best coal. In numerous tests, the relative values of the two fuels have been given thus: 3.75 bbl. of oil is equal to one ton (2000 lb.) of coal; 1 lb. of oil is equal to 1.37 lb. of bituminous coal; 1 lb. of oil is equal to 1.67 lb. of anthracite. The evaporative values of the two fuels has often been given as in the ratio of 11.2 for coal to 15.8 for oil.

From test data in the writer's possession, taken from the same boiler under the same conditions, the evaporative value for run-of-mine Pocahontas coal was 11.77, while for California crude oil it was 16.20. In a test run on a 400-hp. water-tube boiler, equipped with a Hammel burner and backshot furnace, the results obtained are shown in Table 2. The boiler was installed in a power plant in Texas and the test was conducted by the regular plant force.

In comparing these two fuels there are several important factors that must be borne in mind. Their relative evaporative values cannot be based on their calorific values alone. For instance, while theoretically it can be shown that oil requires a greater air supply than coal, in practice it is found that coal, on account of the volatile matter, requires a much greater air supply than oil for complete combustion. The attendant difficulties in the combustion of coal in the furnace, due to its characteristics, is a factor that tends to reduce its evaporative power. On the other hand, the combustion of oil in the furnace is much simpler. The fire can be easily manipulated and a more perfect mixture of air and gases obtained, which tends to make the evaporative qualities of oil superior to those of coal.

Size of Steam Pipes from Chart

By W. L. Durand

SYNOPSIS—For pressures from atmospheric to 200 lb. and capacities from 1 to 10,000 lb. of steam per min., the proper size of pipe may be read directly from a chart.

The most common method used in determining the size of steam pipes is to assume a velocity of 6000 ft. per min. for pipes carrying high-pressure steam, and 4000 ft. for pipes carrying low-pressure or exhaust steam. That this method does not give satisfactory results, except over a very limited range, may easily be seen if the drop in pressure at these velocities is computed for different sizes of pipe. The accompanying table is based on a length of pipe of 100 ft. and an initial pressure of 100 lb. at 6000 ft. per min.

Size of pipe, in.	1	1¼	1½	2	3	4	6	8	10
Loss in pressure, lb.	16	10	7.7	5	2.6	1.7	1	0.68	0.5

From the above, it may be seen that in small pipes the loss in friction is so great as to affect the economy of operation, while, on the other hand, for large pipes, the formula requires a pipe larger than is necessary. This would make the cost of installation excessive since the cost of pipe and fittings increases very rapidly with the size.

The factors which enter into any problem of this character are weight of steam, length of pipe, density of steam, drop in pressure and diameter of pipe. To obtain accurate results, it is necessary to take into account all the above factors and any formula used in this connection should contain all of them and not simply one or two. A good formula, taking into account all of the above fac-

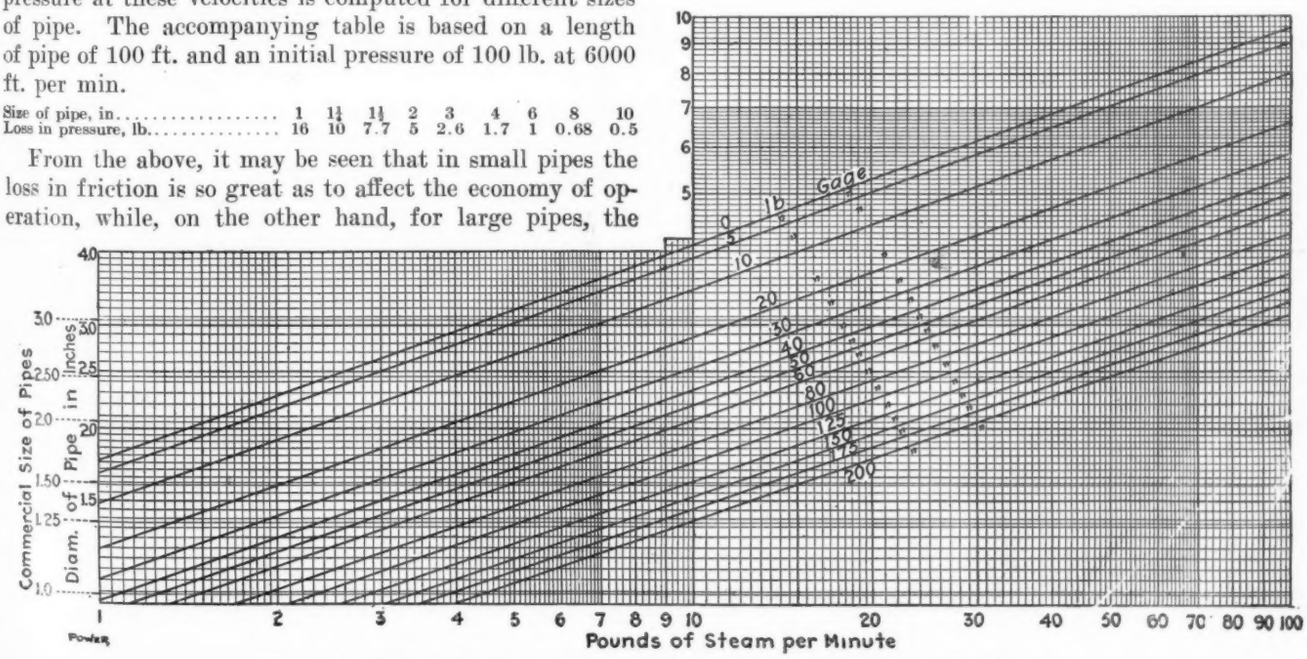


Fig. 1. Size of Pipe up to Capacity of 100 Lb. of Steam per Min.

tors, is the following, given by G. H. Babcock in "Steam."

$$W = 87 \sqrt{\frac{w\,(p_1 - p_2)\,d^5}{L\left(1 + \frac{3.6}{d}\right)}}$$

W = Weight of steam flowing, lb. per min.;
L = Length of pipe, ft.;
d = Diameter of pipe, in.;
w = Mean density, lb. per cu.ft.;
p_1 = Initial pressure, lb. per sq.in.;
p_2 = Final pressure, lb. per sq.in.

The formula given by R. C. Carpenter and Unwin reduce to the same form, except that it has a factor of 87.5.

The reason that this formula is not in more general use is presumably due to the excessive amount of time required in solving it. The writer has been using for some time a set of curves based on this formula with satisfactory results and a great saving in time.

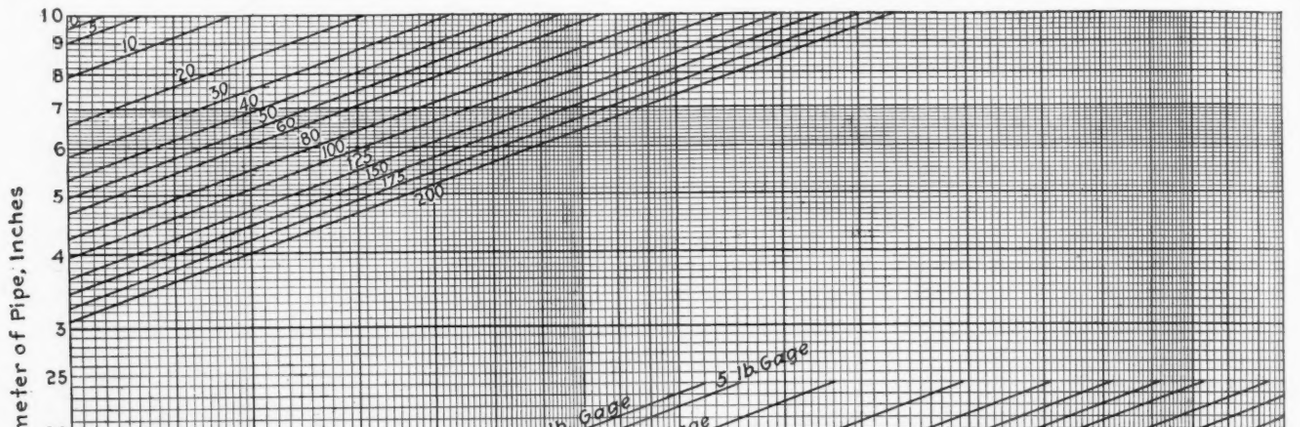


FIG. 2. SIZE OF PIPE FOR CAPACITIES OF 100 TO 10,000 LB. OF STEAM PER MIN.

As there are five variables in the formula and it is possible to plot only three of them without making the curves too complicated, the drop in pressure and length were taken as constants; the length at 100 ft. and the drop as 1 per cent. of the gage pressure, except at 5 lb. and atmospheric pressure, where it is taken as 1 oz. This gives a much better set of curves than if the drop is assumed as a constant at all pressures. If 1-lb. drop is taken, at 200 lb., the drop is too small and the result is a pipe larger than is necessary, with an increase in cost and radiation, while with 5-lb. pressure the loss is excessive.

These curves may be used for any other length or drop in pressure. For any desired length, divide the length expressed in feet by 100 and multiply the weight of steam per minute by the square root of this quotient, and then find the size of pipe for the weight thus found. This will be the proper size for this length of pipe.

For any other drop in pressure, divide 1 per cent. of the pressure by the drop desired and multiply the weight of steam by the square root of this quotient, and then proceed as above.

Take a case where it is desired to transmit 15 lb. of steam per min. at 100 lb. pressure, using the length and drop for which the curves are figured. From the curves a 2-in. pipe will be required. Suppose, however, the length is 250 ft. Proceeding as outlined above,

$$\frac{250}{100} = 2.5;\ \sqrt{2.5} = 1.58;\ 15 \times 1.58 = 23.7$$

From the curves a 2½-in. pipe will be required.

Take another case of 45 lb. of steam per min. at 40 lb. pressure requiring a 4-in. pipe, other factors being as in the problem above. Suppose, however, that it is permissible to have a pressure drop of 1 lb.

$$\frac{0.4}{1} = 0.4;\ \sqrt{0.4} = 0.63;\ 0.63 \times 45 = 28.3$$

This gives a 3½-in. pipe. If a different length and pressure drop is required proceed as above, using the drop per 100 ft.

To determine the length of pipe in inches equivalent to the resistance of one standard 90-deg. elbow, the following formula may be used:

$$L = 76\,d \div \left(1 + \frac{3.6}{d}\right)$$

For a globe valve

$$L = 114\,d \div \left(1 + \frac{3.6}{d}\right)$$

The resistance of gate valves need not be considered.

It might be well to call attention to the fact that, if extreme accuracy is required, the inside commercial diameter of the pipes should be used. These have been marked up to 3 in.

Boiler Explosion near Oil City

BY J. G. DENNINGTON

On the morning of Dec. 13, a 25-hp. locomotive boiler owned by C. P. Irwin and used with a drilling outfit about 12 miles from Oil City, Penn., exploded. No one was killed or injured and the damage was estimated at $300. It appears that the steam line to the engine was

connected to the dome of the boiler by means of a tee. Between this tee and the boiler was a stop-cock and the safety valve was screwed into the top of the tee. Imagine such a connection for the safety valve.

Some repairs to the engine being necessary, the stop-cock was closed so as to shut off all steam from the engine, and when ready to fire up again the operator forgot to open it. The driller in going out to look over the boiler discovered that the gage showed nearly 200 lb. of steam, and remembering that he had forgotten to open the stop-cock started to run to his house near-by to warn his family so they would not be frightened when the boiler let go. The tool dresser followed and just as they got to the house the boiler exploded, scattering over a 10-acre field and breaking nearly all the glass in houses for a half mile around.

The New Massachusetts Institute of Technology Buildings

Plans for the new site of the Massachusetts Institute of Technology on the Charles River Basin, Cambridge, Mass., were recently made public by President R. C. Maclaurin.

Since the fiftieth anniversary of the Institute, some $7,350,000 have been raised for the purchase of the land and the new buildings.

The several sections are clustered about the library which has been made the central feature, with its Roman dome 200 ft. high, rising above a pillared portico. The central court in front of the library is open to the river front and expands near the esplanade to two minor courts. Shrubs, grass plots and fountains are to be used to relieve the severity of the clustered structures; trees are designed to accentuate the corners. The architecture is classical, with pilastered walls, permitting the recesses to be largely of glass. The long facades near the river will have pilasters two stories high and the third story forming the frieze. On buildings farther back a fourth or attic story is added above the entablature. Thus the group rises in height from front to rear. The detail treatment planned is very simple, beauty being sought in magnitude and proportion rather than by ornament. The departmental structures join one another so that it will be unnecessary to go outdoors in passing from one to another. By the use of interior courts, long and narrow wings result, receiving light from both sides.

In planning the position of the several departments, recourse was made to familiar circle diagrams showing the percentage distribution of time to various subjects in the several courses. For instance, half the time is devoted to mathematics and chemistry with the rest in general studies; for the remaining three years the predominant time factors are physics, and electrical and mechanical engineering. General studies and chemistry are housed close together in front of the library, while the electrical-engineering building is just back of it with mechanical-engineering laboratories on one side and physics on the other.

The technical departments converge on the library so far as possible, an arrangement which has been adopted to keep the special libraries of these departments close by the departmental buildings but yet under the care of the library force.

There will be about 2½ acres of drafting rooms, comprising the top floor of all departments except chemistry and general studies. The roofs over these rooms will be broken by standard saw-tooth skylights running across the narrow width of the longer buildings and giving northern exposure but hidden from view below by the high parapets.

The Pratt School of Naval Architecture and Marine Engineering is to occupy the most prominent land position. Only the general features of this structure have been developed, as actual construction must wait a year or two until other conditions have been fulfilled. It has been decided, however, that the school shall have its own entrance from the street; in the basement is to be a towing tank for ship models.

The largest auditorium, seating 2000, is placed within the interior court behind the Pratt School. Parallel with the latter is the mechanical-engineering department, with its heavy laboratories toward railroad tracks in the rear. Back here also is the power plant, of 4000-hp. capacity.

A gymnasium is planned to cost about $100,000, and to have a single floor of 15,000 sq.ft. area and a total floor area of 40,000 sq.ft. At the rear of the grounds is provided space for an athletic field with running tracks and grandstand.

Concerning the "Imperator"

It is but natural that much interest should be taken in such a large and elaborately equipped ship as the "Imperator." It is also to be expected that those interested should wonder why the largest of ocean-going liners should be taken out of service so soon after her maiden voyage. It may be that the daily press, in an effort to satisfy curiosity, has made erroneous statements.

Many rumors have been current. One had it that the boilers were inadequate to furnish steam to get the rated speed, and that oil-burning furnaces were to be used in the hope of remedying the trouble. Then came a rumor that the vessel was structurally weak in parts, followed by statements in the daily press that the builders of the vessel had been ordered by the Hamburg-American Co. to make good, at their own expense, the defects for which they were considered responsible by the owners of the vessel.

Now comes information to the effect that the builders of the "Imperator" guaranteed a speed of 23 knots per hour on 900 tons of coal per day, but that she uses 1150 tons. In consequence of this the turbines are being made over in some respects for the purpose of reducing the steam consumption. In addition to this the vessel is said to be top-heavy and to have been loaded with pig iron in the bottom. This increased the stability of the vessel, but it also increased her immersion. The effect of the pig iron and the vessel's own weight is that even when everything is taken from her that is practicable to remove, she draws 33 ft. 9 in. of water, and about 38 ft. when loaded. On account of this immersion the vessel cannot go behind the break-water at Plymouth, and, therefore, in bad weather it is extremely difficult to land passengers.

The American office of the Hamburg-American Co. states that the home office of the company denies the reports about the inadequate boiler capacity, oil-burning furnaces and the remodeling of the turbines. The reason given for the change in route which avoids Plymouth, is that it facilitates the travel of passengers to land those for Paris at Cherbourg before landing those for the British Isles at Southampton. From Southampton to London is about a two-hour ride while from Cherbourg to Paris takes four hours.

From the time of her maiden voyage until recently laid up, the "Imperator" has been in service seven months and has made seven round trips, carrying a total of about 30,000 passengers. The hot-water system has proved inadequate and changes were badly needed in the crew's quarters. For these two reasons, chiefly, the owners say the vessel was docked for alterations.

Refrigerating Engineers Meet at Dallas

On Dec. 11, 12 and 13 the Practical Refrigerating Engineers' Association held its fourth annual convention at Dallas, Tex., with headquarters at Hotel Adolphus. President Carl Nessler called the convention to order promptly at 10 o'clock, introducing M. S. Church, assistant city attorney, who in behalf of an absent mayor welcomed the association to the city. W. H. Lockett, of Dallas, responded.

After disposing of routine business there was still time at the morning session for the presentation of a paper on "Some Notes on Boiler Attachments," by R. T. Burwell. The paper was written from the standpoint of safety in operation and included some highly practical instruction on the safety valve, water gages and cocks, the steam gage, piping and fit-

tings, the blowoff pipe, the feed pipe and fusible plugs. An extended discussion on feeding the boiler through the blowoff followed, with the consensus of opinion tending toward a safer method.

At the afternoon session J. P. Greenwood read an interesting paper on "Fuel Economy." In plain language the author told how to reduce the fuel bill and discussed the conditions favoring different kinds of fuel used in this part of the country, such as oil, Texas lignite and slack coal.

J. E. Phillips followed with a paper on "Efficiency," pointing out some of the unnecessary and avoidable losses common to the refrigerating plant. The third paper of the session was on "Ammonia" by J. C. Atwood. The properties of ammonia, methods of manufacture, quality of different makes, and incidentally an abbreviated outline of the development of the company with which the author is connected, were all included in the author's remarks.

J. R. O'Shanesy read a paper on "Precooling," briefly outlining the methods that had been employed since 1900 and describing in particular the plant at San Benito, which was ready for service early in 1911. For the benefit of those who could not attend, S. S. Van Der Vaart gave an interesting talk on the Third International Congress of Refrigeration held in Chicago late in September, and next in order was a paper by Joe McCrary entitled "Benefits I Have Derived by the Use of CO_2 Instruments." The author related conditions of the best equipped furnace in the plant when he first began to analyze the gases, and told how the instrument suggested improvements which resulted in a saving of 20 per cent. O. E. Morris talked on the "Relationship between Manager and Chief Engineer," emphasizing the importance of cooperation, mutual understanding and confidence in each other. A discussion on new methods and apparatus for re-icing cars ended the session.

lie in the preparation of the lesson papers. An employment bureau could also be formed to add to the incentive to join the association.

Mr. Van Der Vaart was called upon for an expression, and although he approved of Mr. Kelley's plan, he feared that the organization of chapters in the manner suggested would be impractical at the present time due to the expense attached to the work and very possibly the lack of organizers whose time would be available for the purpose. The same result might be accomplished in a shorter way by affiliating with the National Association of Stationary Engineers. From a talk with Mr. Raven, his association would apparently be willing to take the initiative in organizing chapters in different parts of the United States, these chapters to be affiliated with them, but at the same time members of this body, which will be as separate and distinct as it is today. A local on these lines had been already formed in Chicago.

Members entering into the discussion following were not favorably inclined toward the latter suggestion, as they were afraid of being absorbed by the larger body. It was finally decided to give the committee more time to confer with officials of the National Association of Stationary Engineers and work out a solution of the problem. To aid in the same direction vice-presidents were appointed in every state containing a member of the association.

At the afternoon session the convention listened to some trade talks on Brown's condenser paint, Garland rubber compound to repair leaks and Yale & Towne chain hoists. The time and place for holding the next convention were taken up, and Galveston was the choice of the majority. The following officers were elected. A. E. Page, president; D. L. Gilbert, Texas, first vice-president; Robert Jeffreys, Louisiana, second vice-president; Guy Conley, Arkansas, third vice-president; W. L. Storey, Oklahoma, fourth vice-president; J.

DELEGATES TO THE FOURTH ANNUAL CONVENTION OF THE PRACTICAL

In the evening a smoker and Dutch lunch in the N. A. S. E. Hall proved to be a most enjoyable affair. The "get together and get acquainted" spirit predominated and Southern hospitality was in evidence.

On Friday morning a short account was given of the "Burhorn" cooling tower exhibited at the Third International Congress of Refrigeration, and the remarkable results obtained in a closed building with no movement of air. Brief discussion on this subject was followed by an extended controversy involving the policy and, as considered by some of the members, the life of the association.

A letter from H. H. Kelley, editor of the "Southern Engineer" and a member of the committee elected a year previous to provide a plan for the expansion of the association, was read at the meeting. Mr. Kelley outlined a plan calling for the formation of chapters, which should be controlled by a central or governing board. This board would be the national head of the association and composed of delegates selected by the several local bodies. The national body would hold annual meetings at various points as the one body does now. The money from dues would be spent for educational purposes, dispensed through the governing body and the education might comprise a thorough graded course in refrigerating engineering and subcourses in allied lines. The courses would be similar to those offered by the correspondence schools, but as the lessons would be studied jointly in the locals, the expense for postage between them and the national body would be small. The principal expense would

H. Higginbotham, Mississippi, fifth vice-president; W. W. Hughes, Georgia, sixth vice-president; Sam Bayless, Alabama, seventh vice-president; W. A. Owen, Tennessee, eighth vice-president; F. H. Ladd, Florida, ninth vice-president; W. H. Beisley, South Carolina, tenth vice-president; Thyson Bernard, Virginia, eleventh vice-president; O. E. Morris, secretary-treasurer; R. G. Frazer, director; C. W. McGiboney, examiner.

Saturday was spent in visiting the North Fort Worth Stock yards upon the invitation of Armour & Co., the Texas Brewing Co., of Fort Worth and other plants of interest.

Long Acre Co. Threatens Damage Suit

Following the recent court decision sustaining the franchise of the Long Acre Electric Light & Power Co., it is reported that this company intends bringing suit against the New York Edison Co., the A. N. Brady Estate and the Consolidated Gas Co. for alleged damages due to loss of profits during the period that its business has been tied up because of legal opposition from these interests.

Reviewing briefly the history of the case, the Long Acre Electric Light & Power Co. was organized in 1906 and steps were immediately taken to establish its right to use the conduits of the Consolidated Subway & Telegraph Co. for run-

ning its wires. This privilege was affirmed by the courts in 1907.

One year later the company applied to the Public Service Commission for permission to issue $50,000,000 in bonds and $10,000,000 in stock. This application was denied but later the commission voted to allow $2,000,000 in stock and $4,000,000 in bonds.

The New York Edison Co., after protesting to the commission, appealed from its decision, and when it lost in the Appellate Division, the case was carried to the Court of Appeals, which, on Dec. 19, sustained the action of the Public Service Commission.

Recent Court Decisions

Digested by A. L. H. STREET

Right of Way for Oregon Electric Lines—The authority given under the statutes of Oregon to corporations to construct lines along streets and highways to convey electric power, etc., extends to corporations doing business in the state, though organized under the laws of another state. (Oregon Supreme Court, Northwestern Electric Co. vs. Zimmerman, 135 Pacific Reporter, 330.)

Abandonment of Water Rights—A water right is not abandoned by mere nonuse of it; abandonment will be declared only when shown to have been clearly intended by the possessor of the right. (Washington Supreme Court, Thorp vs. McBride, 135 Pacific Reporter, 228.)

Conveyance and Notice of Water Privileges—A deed, in ordinary form, to mill property, including "all and singular, the tenements, hereditaments and appurtenances thereto belonging or in any wise pertaining," carries with it water privileges appurtenant to the property, without special mention of those privileges. One purchasing land under a deed, which refers to a map on file in the county clerk's office, which map indicates the presence of a dam owned by a person owning land on the other side of the stream, is chargeable with constructive notice of the rights held by the dam owner. (New York Supreme Court, Monroe County Equity Term; Barnes vs. Martin, 143 New York Supplement, 792.)

When City is Estopped to Declare Forfeiture of Power Franchise—A city which has granted a franchise to a company to produce and sell light and power, will not be permitted by the courts to dispute the validity of the franchise on the ground that, before it was granted, the company failed to procure the consent of abutting property owners to the placing of poles and stringing wires in the streets, as required by the laws of Illinois, where, acting on the faith of the franchise, the company has made large improvements and incurred much expense. "If the defendant had been wrongfully occupying the strets, the public authorities, in the exercise of proper diligence to prevent encroachment on the public right, might have compelled the cessation of such occupation. When they not only failed to do so, but took affirmative action with the apparent approval of every one interested, under which the situation was changed, extensive improvements were made, and large liabilities were incurred, the principles of equity require that the city should be estopped." (Illinois Supreme Court, People vs. Union Gas & Electric Co., 103 Northeastern Reporter 245).

NEW PUBLICATIONS

AN EXTENSION OF THE DEWEY DECIMAL SYSTEM OF CLASSIFICATION APPLIED TO THE ENGINEERING INDUSTRIES. By L. P. Breckenridge and G. A. Goodenough. Originally issued as Bulletin No. 9, in 1906, and still published by the Engineering Experiment Station, University of Illinois, Urbana, Ill. Price, 50c.

The filing and classification of engineering data has become a matter of much importance, and this bulletin was prepared for use as a guide in carrying out such work. The original edition of this bulletin was subject to the usual gratuitous distribution and the demand was such that a second edition was printed and distributed. The demand continuing, a revised edition, much extended as compared with the original eition, was printed in the fall of 1912. This edition was limited and is not subject to gratuitous distribution. It presents subdivisions of subjects in such detail as to constitute a complete classification for most engineering industries, even though they are highly specialized. The revision has been made in accordance with the 1911 edition of "Decimal Classification" by Melvil Dewey.

BOOKS RECEIVED

VACUUM-CLEANING SYSTEMS. By M. S. Cooley. Heating and Ventilating Magazine Co., New York. Cloth; 232 pages, 6¼x9½ in.; 104 illustrations; tables. Price, $3.

REFRIGERATING ENGINEERS' ASSOCIATION AT DALLAS, TEX.

AMERICAN ELECTRICIANS' HANDBOOK. By Terrell Croft. McGraw-Hill Book Co., New York. Leather; 711 pages, 4¼x7 in.; illustrated; tables. Price, $3.

HANDBOOK FOR MACHINE DESIGNERS AND DRAFTSMEN. By Frederick A. Halsey. McGraw-Hill Book Co., New York. Cloth; 494 pages, 8½x11¼ in.; illustrated; tables. Price, $5.

COMPRESSED-AIR PRACTICE. By Frank Richards. McGraw-Hill Book Co., New York. Cloth; 326 pages, 6x9¼ in.; 96 illustrations; tables. Price, $3.

SHOP SKETCHING. By J. W. Woolley and R. B. Meredith. McGraw-Hill Book Co., New York. Cloth; 102 pages, 6x9¼ in.; 122 illustrations. Price, $1.

MECHANICAL REFRIGERATION. By H. J. Macintire. John Wiley & Sons, New York. Cloth; 346 pages, 5¾x9 in.; 121 illustrations; plates; tables. Price, $4.

QUESTIONS AND ANSWERS RELATING TO MODERN AUTOMOBILE DESIGN, CONSTRUCTION AND REPAIR. By Victor W. Page. The Norman W. Henley Publishing Co., New York. Cloth; 622 pages, 5x7½ in.; 329 illustrations; plates. Price, $1.50.

TRANSFORMER PRACTICE. By William T. Taylor. McGraw-Hill Book Co., New York. Second edition. Cloth; 278 pages, 6¼x9½ in.; 191 illustrations. Price, $2.50.

Trade School Paper—"Occasional Suggestions" is the name of a little paper being published to present the principles underlying the methods of the Worcester Boys Trade School. The first issue, a small eight-page paper of a size convenient for the pocket, contains helpful suggestions for those who are interested in the training of mechanics.

SOCIETY NOTES

The Harlem Y. M. C. A., New York City, will, on Jan. 6, offer a complete course to those interested in heating and ventilation. It will consist of lectures, demonstrations of systems and appliances, and discussion. The instructor is C. A. Fuller, a well known heating engineer. All information will be freely given by application to the Educational Department, Harlem Y. M. C. A., 5 West 125th St., New York City.

PERSONALS

L. E. Collins, formerly with Yeomans Bros., has associated himself with the Western Pump & Engineering Co., Chicago, and will specialize in house-service and bilge pumps.

A. R. Maujer, western editor of "Power," gave an illustrated lecture, Dec. 20, on chimneys at the joint meeting of the Milwaukee subordinate associations of the National Association of Stationary Engineers.

CIVIL SERVICE OPPORTUNITIES

Situation for Hoisting Engineer—The Chicago Civil Service Commission will hold an examination Jan. 8, for hoisting engineer, class Ea, grade II. The subjects of examination will be: Special subject, experience, arithmetic and report.

Situations for Mechanical Draftsmen—The Chicago Civil Service Commission will hold an examination, Jan. 15, for the position of mechanical engineering draftsman, class B, grade II. Salary, $1500 to $1740. Subjects of examination: Special subject, mathematics, experience and report. Jan. 19 an examination will be held for mechanical draftsman, class B, grade I. Salary $1080 to $1320. Subjects of examination: Special subject, mathematics, experience and report.

Situation for Expert in Fire Prevention—The New York State Civil Service Commission will hold an examination for the foregoing position, Jan. 17, 1914. Salary, $3500. Candidates must have had at least ten year's experience in solving problems involved in direct and abutting fire hazards; the means and adequacy of exits from buildings; the interior arrangement and equipment of factories from the standpoint of safety of persons therein in case of fire or panic; the design, arrangement, installation and use of appliances for fire prevention. An appointee is desired between 30 and 60 years old. He must be a citizen of the United States and a resident of New York State. Application blanks should be sent for before Jan. 7, and returned by Jan. 9 to the secretary of the Civil Service Commission, Albany, N. Y.

Situation for Assistant Inspector of Gas—Public Service Commission, Second District. The New York State Civil Service Commission will hold an examination for the foregoing position Jan. 17, 1914. Salary, $1080 to $1200. The duties are to inspect and test gas for its purity and quality. Necessary traveling expenses will be paid in addition to the stated salary. Candidates must be college graduates in chemistry or have an equivalent education. Subjects of examination and relative weights: General and theoretical chemistry, quantitative analysis, photometry, calorimetry and manufacture of gas, 7; education, experience and special training, 3. Those desiring to compete should send for application blanks before Jan. 7, and return them by Jan. 9.

Situation for Chemical Engineer—The New York State Civil Service Commission will hold an examination for the foregoing position, Jan. 17, 1914. Salary, $3500. The duties require a knowledge of chemical engineering to determine the dangers to life and health arising from chemical manufacturing processes, and for the proper solution of similar problems. An appointee is desired between the ages of 30 and 60, who has completed a satisfactory course in chemical engineering, or other technical course, and who, in addition, has had practical experience in his profession or in work related to the duties of the position in question. Candidates must be citizens of the United States and residents of New York State. Application blanks should be sent for before Jan. 7 and returned before Jan. 9 to the Secretary of the Civil Service Commission, Albany, N. Y.

Situation for Laboratory Assistant—The United States Civil Service Commission announces an examination for laboratory asistant, men only, on Jan. 21 and 22, 1914, to fill vacancies in this position in the Bureau of Standards, Department of Commerce, Washington, D. C., at salaries ranging from $900 to $1200 a year. The work is similar to that in the better class of industrial and scientific research laboratories. Competitors will be examined in general physics, French or German, practical questions on each optional subject chosen, education, training and experience. The following are the optional subjects: Electricity, electrical engineering, heat, mechanics, mechanical and civil engineering, optics, paper technology and textile technology. An educational training equivalent to graduation from a college or university of recognized standing is required. Applicants must be between 20 and 35 years old and citizens of the United States. Persons who desire this examination should at once apply for application Form 1312 to the United States Civil Service Commission, Washington, D. C.

Situation for Assistant Steam Engineer, State and County Hospitals, Departments and Institutions. The New York State Civil Service Commission will hold an examination for the foregoing position, Jan. 17, 1914. Usual salaries, $480 to $600 with maintenance when employed in institutions. Candidates, unless technical graduates, must have had two years' practical experience as machinist, boiler maker, steamfitter, oiler, fireman, engineer or electrician, including at least one year in the operation of stationary engines or electrical machinery. Technical graduates will be required to have had at least one year's satisfactory experience since graduation. Subjects of examination and relative weights: Practical and technical questions on the operation and repair of steam engines, boilers, steam-heating apparatus and electrical machinery, 6; experience, education and personal qualifications, 4. Those desiring to compete should send for application blanks before Jan. 7, and return them by Jan. 9, addressing the secretary of the Civil Service Commission, at Albany, N. Y. Candidates must be citizens of the United States and residents of New York State.

Situation for Assistant Electrical Engineer, state hospitals, departments and institutions. The New York State Civil Service Commission will hold an examination for the foregoing position Jan. 17, 1914. Usual salaries, $480 to $900 and maintenance. Candidates who are graduates in mechanical or electrical engineering must have had one year's practical experience in electrical work, or in the operation of stationary steam engines and electrical machinery. Other candidates must have had two years' practical experience as engineer or electrician, including at least one year in the operation of stationary engines or electrical machinery. Subjects of examination and relative weights: Practical and technical questions on electricity, electrical machinery and apparatus, steam engines, and steam and electric-power house practice, etc., 6; experience and education, 4. Those desiring to compete should send for application blanks before Jan. 7, and return them by Jan. 9, addressing the secretary of the Civil Service Commission at Albany, N. Y. Candidates must be citizens of the United States and residents of New York State.

Situation for Electrical Engineer (Assistant)—Public Service Commission, First District. The New York State Civil Service Commission will hold an examination for the foregoing position Jan. 17, 1914. Salary, $1801 to $2100. Appointees to this position will have to handle considerable work in connection with theoretical computations on the design of electric railroad equipment, and should be conversant with and experienced in the design, selection, installation and inspection of electric railroad motors, signals, switching apparatus, etc. Applicants must have had at least seven years' engineering experience in electric railroad work preferably with rapid transit or heavy electric traction lines. Graduation in engineering from a school of the highest grade will be accepted as equivalent to four years' experience. Subjects of examination and relative weights: Written examination, 2; experience, training and personal qualifications, 3. In rating the last mentioned subject an oral examination will be held. Those desiring to compete should send for application blanks before Jan. 7, and return them by Jan. 9, addressing the secretary of the Civil Service Commission of Albany, N. Y. Candidates must be citizens of the United States.